AN EXPERIMENTAL FACILITY FOR DYNAMIC COMBINED ENVIRONMENTS RELIABILITY TESTING OF AIRBORNE EQUIPMENT

AIR FORCE FLIGHT DYNAMICS LABORATORY
WRIGHT-PATTERSON AIR FORCE BASE, OHIO

OCTOBER 1976

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AN EXPERIMENTAL FACILITY FOR DYNAMIC COMBINED ENVIRONMENTS RELIABILITY TESTING OF AIRBORNE EQUIPMENT

ENVIRONMENTAL CONTROL BRANCH
VEHICLE EQUIPMENT DIVISION

OCTOBER 1976

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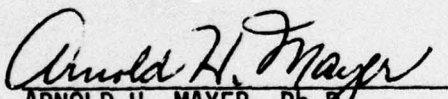Approved for public release; distribution unlimited

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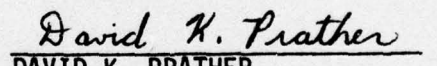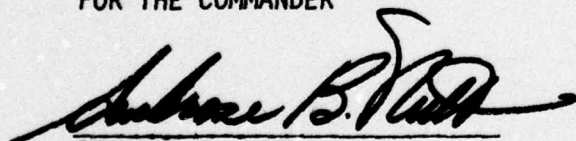WRIGHT-PATTERSON AIR FORCE BASE, OHIO 45433

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A newly developed experimental facility can assess in-service reliability of airborne equipment under the dynamic combined environments encountered in flight. Virtually complete simulation of the airborne equipment environment is achieved by predominantly magnetic tape controlled scheduling of random mechanical vibrations, compartment air temperature, altitude, cooling air flow rate, temperature, and humidity at the rapid rates of variation produced in high performance aircraft.

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The design, engineering development, operational capability, and associated
aspects of construction and operation are documented along with some early
operating experience with actual avionics hardware.

FOREWORD

This report was prepared by the Combined Environments Test and
Thermal Control Groups, Environmental Control Branch, Vehicle Equipment
Division, Air Force Flight Dynamics Laboratory. The work described was
conducted under Project 6146, Task 61460413.

The authors express their appreciation to Carl Williams and Herbert
Knick, AFFDL/FEE, for their substantive contribution to chamber modifi-
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C. Savage for the opportunity to work on this project.

The performance period for the work was from January 1974 through
October 1975. This report was submitted December 1975 by the authors.

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NOMENCLATURE AND SYMBOLS

Symbol	Definition
a	Constant
A	Function
b	Constant
C	Specific Heat
C_p	Specific Heat of Air at Constant Pressure
C_v	Valve Flow Coefficient
E	Controller Output Voltage
f(TDEW)	Hygrometer Output Function
g, G	Gravitational Acceleration, or a Constant Defined in Text
HX	Heat Exchanger
K_1, K_2, K_3	Controller Function Weighting Factors
L	Length (ft)
LN_2	Liquid Nitrogen
M	Mass
P	Pressure (psia)
PV	Process Variable
Q	Volumetric Flow Rate
R	Ideal Gas Constant
S	Laplace Transform
SP	Setpoint
T	Temperature (°F or °R)
TDEW	Dewpoint Temperature (°F)
t	Time (sec or min)

NOMENCLATURE AND SYMBOLS (Concluded)

Symbol	Definition
V	Volume
V_p	Controller Input Voltage for Pressure
V_T	Controller Input Voltage for Temperature
V_{TDEW}	Controller Input Voltage for Dewpoint Temperature
V_W	Controller Input Voltage for Flow Rate
W	Flow Rate (lb/min)
X	Variable
τ	Time Constant (sec)
ρ	Density (slug/ft^3)
ΔT	Temperature Difference (°F)
ΔP	Pressure Difference (psi)
ϵ	Heat Exchanger Effectiveness
γ	Constant
ζ	Variable

SUMMARY

An experimental facility has been developed for assessing the re-
liability of airborne equipment under the dynamic combined environments
encountered in flight. Realistic simulation of the dynamic air-cooled
equipment environment has been achieved by automatically scheduling
random mechanical vibration, compartment temperature, altitude, cooling
airflow rate, temperature and humidity at the rapid rates of variation
induced in military high performance aircraft. All environmental param-
eters except compartment temperature and mechanical vibrations are auto-
matically controlled by a preprogrammed magnetic tape permitting around-
the-clock operation with a single human monitor in attendance.

A pre-existing (4 ft diameter x 5 ft long) temperature-altitude-
vibration test chamber provided the nucleus for the new dynamic through-
flow facility. A cam-operated compartment temperature controller and
the vibration capability were retained unmodified. The altitude control
was redesigned to provide the rapid rate of response required under
through-flow conditions.

Cleaned and dried compressed air is used to provide process and
control air for the facility. Pneumatically controlled valves meter the
airflow, provide thermal conditioning by proportioning the airflow be-
tween hot and cold heat exchangers, and adjust humidity by controlling
steam injection into the airstream.

A performance summary of the dynamic environments simulator over appropriate parameter ranges and corresponding rates of change is given in the following table:

Environmental Parameter		Range	Maximum Rate	Conditions
1. Chamber Temperature	(1)	-100 to 300°F	±10°F/min	
2. Chamber Altitude	(1)	0 to 66,500 ft	66,500 ft/min	0 lb/min
	(2)	0 to 44,000 ft	44,000 ft/min	3.2 lb/min
	(3)	0 to 38,000 ft	38,000 ft/min	5.7 lb/min
3. Equipment Vibration				
Sine or Random	(1)	5 to 2,000 Hz		
Max Force	(2)	8,000 LBF		
4. Cooling Airflow Rate	(1)	0 to 5.7 lb/min	170 lb/min/min	
5. Cooling Capacity	(1)	0 to 1.5 KW		
6. Cooling Air	(1)	-90 to 200°F	0°F/min	0 lb/min
Temperature	(2)	255°F ΔT	10°F/min	1.6 lb/min
	(3)	190°F ΔT	100°F/min	5.7 lb/min
	(4)	120°F ΔT	100°F/min	1.6 lb/min
7. Cooling Air Dewpoint	(1)	-20°F to 92°F	134°F/min	{ 5.7 lb/min, 14.4 psia
	(2)	16°F to 92°F	91°F/min	{ 1.6 lb/min, 14.4 psia

Initial operation involving combined environments reliability testing (CERT) of an airborne radar system in the experimental facility while using appropriate environmental profiles is briefly described. A largely faithful reproduction of field failure modes and rates was exhibited over the 670 hours of test duration. Greater detail is provided in a companion report devoted exclusively to this subject.

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SECTION I

INTRODUCTION

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1. PURPOSE OF THE REPORT

This report documents the design, development, operation, and capabilities of the first experimental, dynamic, combined environments simulator for reliability testing of forced-air-cooled airborne equipment. The need for and usefulness of a Combined Environments Reliability Test (CERT) facility is discussed in paragraph 2 of this introduction. Section II discusses the basis and derivation of the experimental CERT facility design requirements, and describes the schematic and quantitative design of a first experimental version of such a facility. The transient performance capability of the CERT facility is treated in Section III where it is shown to satisfy the basic design specifications. A brief account of initial operation experience is included in Section IV. Conclusions are provided in Section V and recommendations regarding design of the next generation of CERT facilities are made in Section VI.

2. NEED FOR A DYNAMIC COMBINED ENVIRONMENTS RELIABILITY TEST FACILITY

In order to increase its effectiveness, the modern military aircraft has become a heavy user of electronics equipment. As a consequence, airborne avionics now represent 20 to 30% of the total cost of a weapon system (Reference 1). Furthermore, the trend in expenditures by the Defense Department for avionics over the last ten years has been one of exponential growth (Reference 1).

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The reliability of avionics equipment has a direct impact on the effectiveness of an air weapon system, through its influence on the probability of mission success, and on its life cycle cost to the extent that it affects required system redundancy, stock levels of replacement units, and cost of maintenance actions. In order to reach the objectives of increased weapon system effectiveness and reduced life cycle costs, it is imperative that cost-effective means exist for verifying, improving, and predicting (or assessing) the reliability of avionics equipments within their service environment well in advance of actual commissioning.

Reliability verification is a step of the acquisition process in which there exists a need to establish that procured avionics equipments satisfy the reliability levels required by their procurement specifications. Reliability improvement is required during the engineering development of avionics equipment when those design features accounting for low reliability must be identified and subsequently corrected by redesign. A means of prediction or advance assessment of reliability is required by weapon systems planners such as operations analysts and logistics planners.

Reliability is a property highly specific to a given type of equipment in a particular service environment. It is a fact that the practice of using reliability data pertaining to one type of equipment in its environment toward predicting reliability when either the equipment type or its environment are different is notably unreliable (Reference 2). Field failure rates usually fall between three and eight times the

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values estimated or predicted following MIL-HDBK-217B procedures (References 3 and 4).

Concerning MIL-HDBK-217B, it should be pointed out that the procedures recommended therein are not claimed to provide absolute values of reliability but are useful for comparative purposes and for identifying reliability problem areas. Steady-state operating temperature is the prime environment to which failure rates are continuously related. Additional overall factors account in a coarse fashion for aspects of component quality and broad applications categories such as space, aircraft, and inhabited, uninhabited, pressurized and unpressurized compartments.

The conspicuous lack of success of analytical techniques to accurately predict avionic equipment failure rates is due, in most cases, to a number of causes. Chief among these is the lack of a sufficiently comprehensive and refined data base and a methodology to relate failure rates to the mixes and frequencies of combined stresses experienced by equipment components. It is highly probably that such a data base will never be available. Even given the data base, the labor involved in calculating combined stresses and incidence frequencies associated with each component in a large system appears prohibitive and would not be carried out to the accuracy required by operations analysts and logistics planners. Added to these considerations is the modifying influence of human factors as they enter through equipment maintenance and reporting practices and procedures which have been suggested as another difficult-to-quantify cause of the unpredictability of avionic equipment

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reliability. When analytical techniques fail to provide the required confidence, testing is usually resorted to. It is readily seen, therefore, that a testing capability that subjects avionics hardware to simulated patterns of in-service combined environments in order to determine reliability in a particular application is the most effective tool for verifying, improving and assessing avionics equipment reliability.

The concept of combined environments testing for avionics equipment has been advocated and, to a modest extent, applied for a number of years. However, current combined environments test methods achieve only limited success in predicting field performance. A comparison of existing test and flight environmental conditions reveals that the dynamic aspects of flight combined environments are not simulated. This has prompted environmental and reliability specialists to begin developing test criteria which can be structured effectively to "fly" avionics equipment in the laboratory.

This approach, commonly referred to as "environmental profile" or "flight profile" testing, is directed at establishing combined environments test criteria which include the significant environmental parameters affecting field performance and reliability. The parameters cited as most significant are temperature, altitude, humidity and vibration. A recent study (Reference 5) concluded that these parameters account for 90% of all environmentally induced avionics field failures.

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Although temperature, altitude, humidity and vibration combined environments test facilities are currently available, the trend toward increased use of forced-air-cooled avionics equipment makes these facilities obsolete. In forced-air-cooled avionics systems conditioned air from the aircraft's Environmental Control System (ECS) is blown directly through the equipment or through cold plates to maintain electronic component design temperatures. In-flight measurements of conditioned air temperature, flow rate and humidity show rapid variations in these parameters during a mission profile. Several studies (References 5 and 6) indicate that cycling of conditioned air parameters seriously degrades avionics performance and reliability and should be included in environmental test procedures.

The implementation of flight profile testing for forced-air-cooled avionics equipment requires a test facility that can duplicate the dynamic characteristics of in-flight forced-air cooling in addition to that of equipment bay temperature, altitude and vibration. The design, fabrication, and performance of such a facility are discussed in this report.

SECTION II

DESIGN AND DEVELOPMENT OF EXPERIMENTAL CERT FACILITY

1. PROGRAM OBJECTIVES AND APPROACH

Recognizing the important role that a CERT facility could play in evaluating avionics reliability, a decision was made to undertake development of an experimental CERT facility as a means of assessing the feasibility of such a tool.

Design criteria stipulated at the outset were simplicity, low construction and operating costs, maximum use of existing and commercially available equipment, and short development time. Development was to consist of two phases. An initial design, development, construction and checkout phase on manual operation of the facility was to be followed by a second phase in which capability for fully automatic operation would be provided. The automatic setpoint control system for the facility was to permit flexible, automatic scheduling of environmental profiles to accommodate frequent profile changes and around-the-clock testing with minimum requirements for human operators. Performance specifications for the facility were derived from an analysis of measured in-flight environmental data taken from aircraft compartments.

The goal of maximum use of existing equipment was addressed by selection of an available static temperature-altitude-vibration test chamber as the nucleus for the complete, dynamic, combined environments

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facility with its through-flow capability. Incorporation of an existing air compressor to pressurize the process and control air constituted another example of advantageously utilizing existing equipment.

2. DEFINITION OF DESIGN SPECIFICATIONS

Specifications for facility performance were determined as follows:

(1) An analysis of Category II Climatic Flight Test Reports (References 7, 8 and 9) was conducted to determine both the parameter ranges and rates of change for compartment static air temperature, pressure/altitude, cooling airflow rate, cooling air temperature and its humidity.

The test reports contain measurements of temperature and pressure parameters throughout the aircraft equipment bays and environmental control system, along with aircraft mission profile parameters, i.e., altitude, mach number, and engine speed. Since cooling airflow rate and humidity measurements do not appear in Category II reports, these parameters had to be deduced from the existing measurements.

The calculation procedure for airflow utilized a quasi-steady, Fliegner-type (Reference 10) correlation for cooling turbine flow in terms of inlet pressure, temperature, nozzle area, and a nominal constant value of the flow coefficient. The dewpoint of the cooling air was taken as the lesser of ambient and turbine exit temperatures. Cooling air delivery temperature was taken equal to cooling turbine discharge temperature. It may be noted that the foregoing approach is conservative from the standpoint of CERT facility design.

(2) Flight vibration data derived from recent analyses (Reference 11) were applied to specify the frequency bandwidth, center frequency, the random, periodic or shock nature of the excitation, and displacement amplitudes of mechanical equipment vibrations.

Typical environmental profiles which had to be simulated are illustrated in Figure 1. Representative measured avionics flight vibration data is shown in Figure 2.

The facility performance specifications derived from analysis of measured flight compartment environmental data are presented below.

Parameter	Range	Rate of Change
Compartment Temperature	-65 to 250°F	±30°F/min
Altitude	0 to 60,000 ft	±20,000 ft/min
Airflow Rate	0 to 100% (0 to 6 lb/min)	±80%/min (±4.8 lb/min/min)
Airflow Temperature	-65 to 120°F	
@ max airflow		±80°F/min
Airflow Humidity	1.5×10^{-5} to 0.032 lb H_2O/ lb air (-65 to 90°F dewpoint)	±0.03 lb H_2O/lb air/min (±80°F/min dewpoint)
Vibration	5 to 2,000 Hz	0.1 G^2_{rms}/Hz Random 15 G_{peak} Sinusoidal

The flow specification was also given in percent of maximum flow to permit application to any size facility. Maximum flow is determined by comparing the heat dissipation range of projected avionics test items

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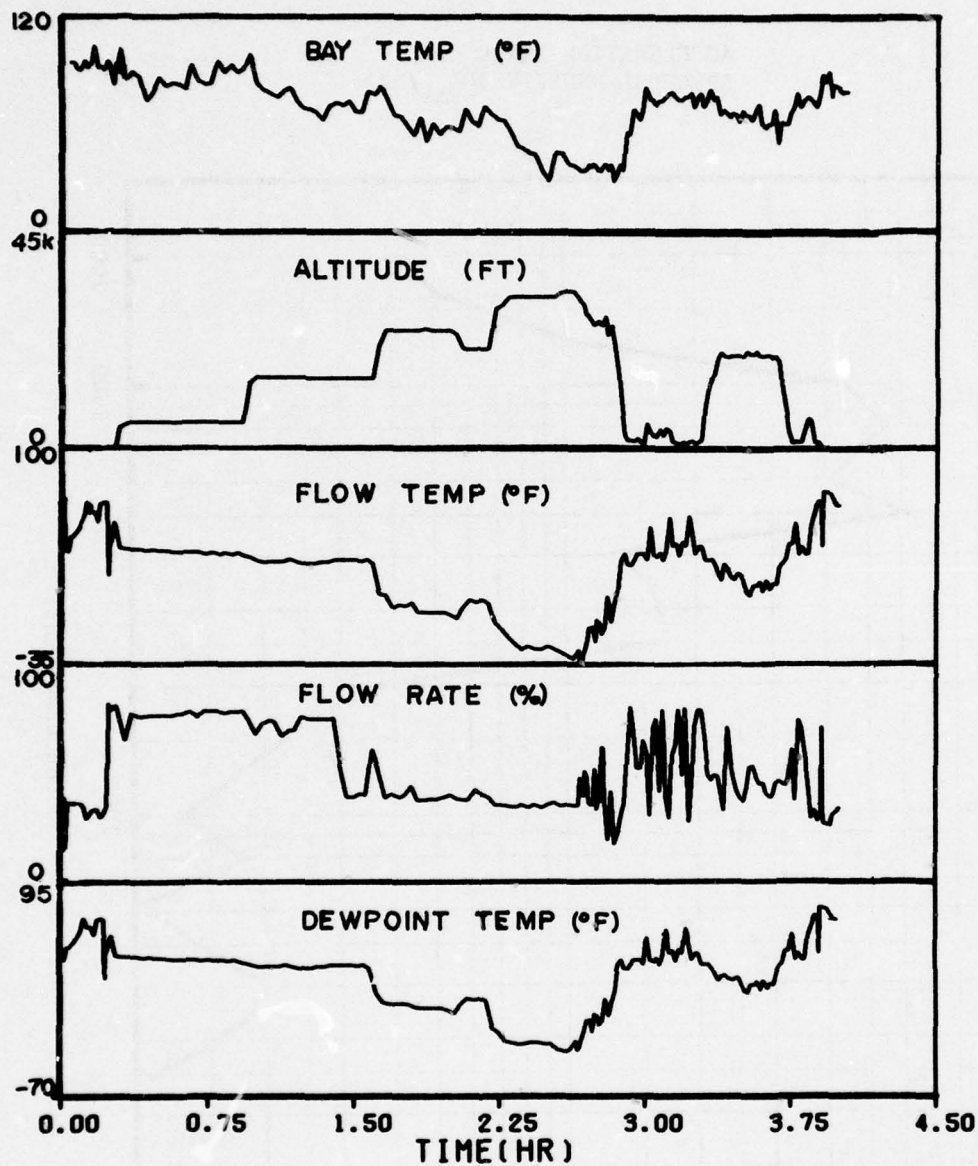
Figure 1. Example of Environmental Profiles of a High
Performance Aircraft in a Tropic Climate

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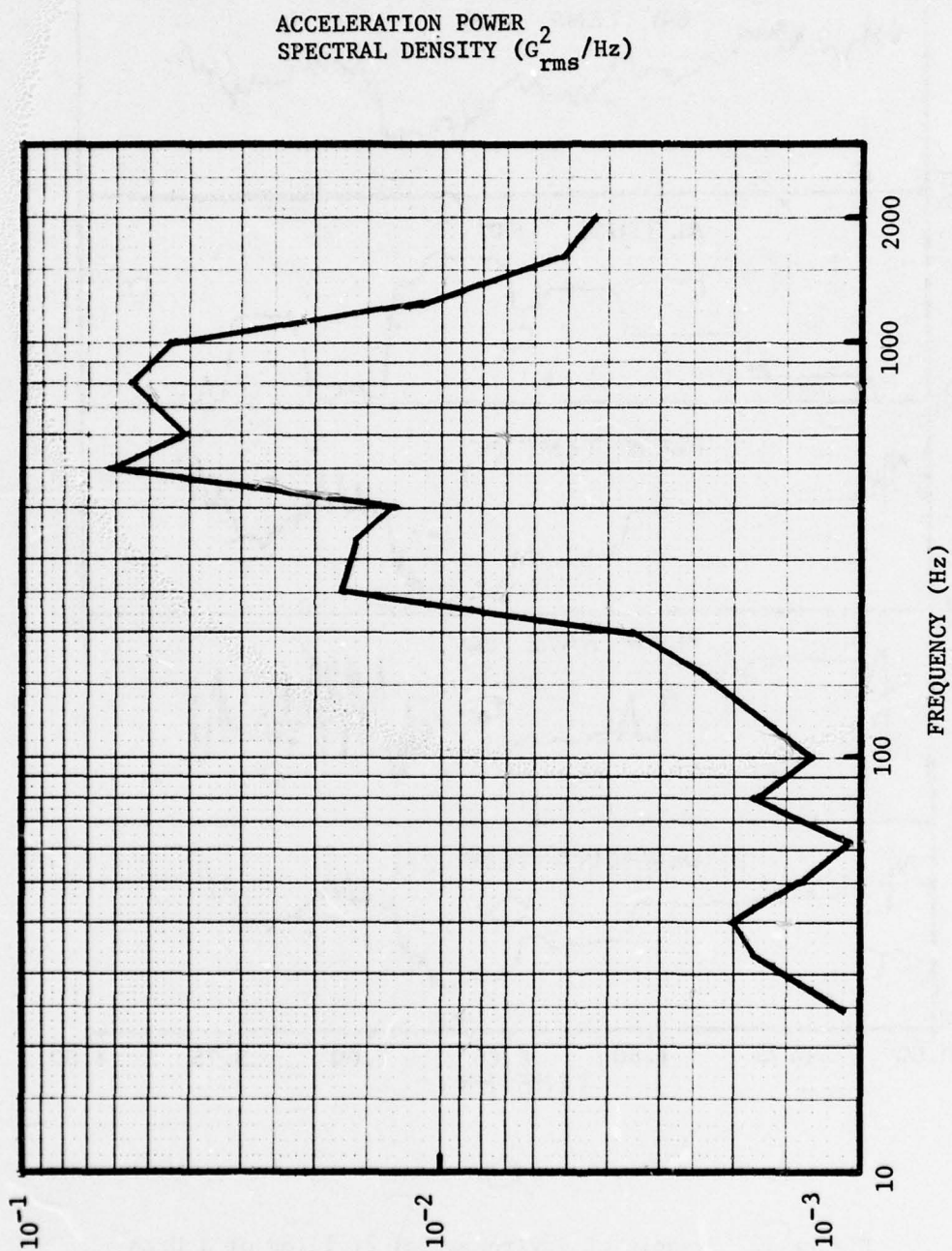
Figure 2. Example of Avionics Equipment Vibration in a High Performance Aircraft

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with general avionics cooling requirements shown in Figure 3. The maxi-
mum flow rate selected for the experimental CERT facility was 6 lb/min,
which is nominally adequate for a 1.5 kilowatt avionics dissipation load
at 80°F as seen in Figure 3 (Reference 12). The three curves (A, B, C)
apply respectively to high watt density micro-electronic equipment,
medium watt density solid-state electronics, and low watt density non-
solid-state electrical equipment installed in a modern high performance
aircraft.

3. DESIGN OF THE CERT FACILITY

3.1 OVERVIEW OF FACILITY

The open loop arrangement adopted to satisfy the facility design
goals listed in paragraph 2 is pictured in Figure 4. In this concept,
conditioned process air is supplied to a test chamber at independently
scheduled flow rate, temperature and humidity while the chamber under-
goes an equally independent program of changes in pressure (altitude)
and static air temperature. Process air taken from the atmosphere is
subjected sequentially to pressurization, clean-up, flow control, ther-
mal conditioning and humidification processes, respectively, before in-
troduction to the temperature-altitude-vibration chamber containing the
equipment undergoing test. A shaped spectrum of mechanical vibrations
is applied to equipment mounted within the chamber by an electromagnetic
vibrator located externally.

3.2 TEST CHAMBER

Maximum use of existing equipment was made by incorporating an
existing temperature-altitude-vibration chamber into the expanded
capability for dynamic simulation of altitude, airflow rate, airflow

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A. > 1 watt/in^2
B. = .3 watt/in^2
C. < .1 watt/in^2
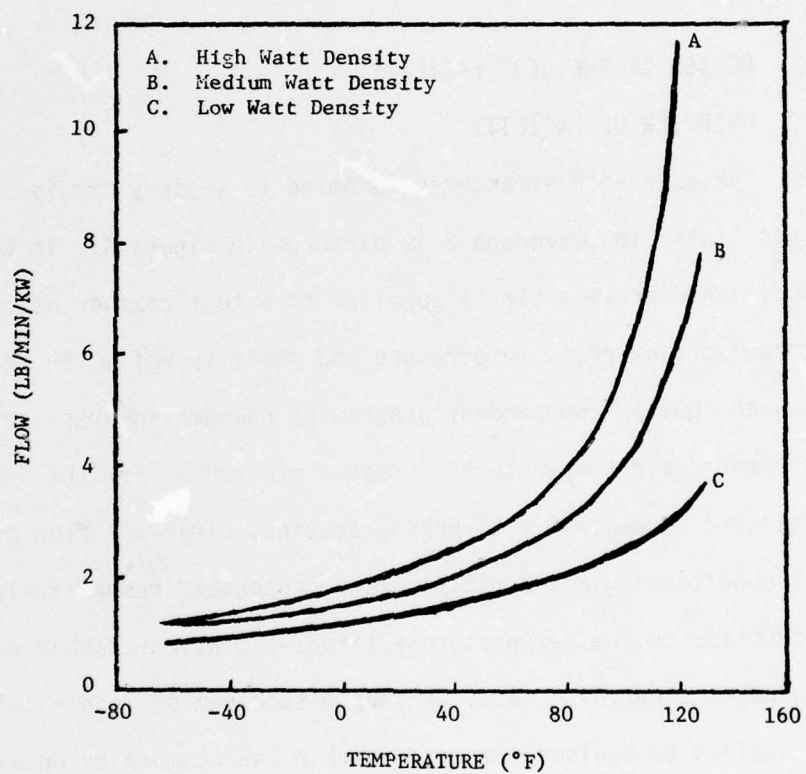
Figure 3. Avionics Cooling Requirements

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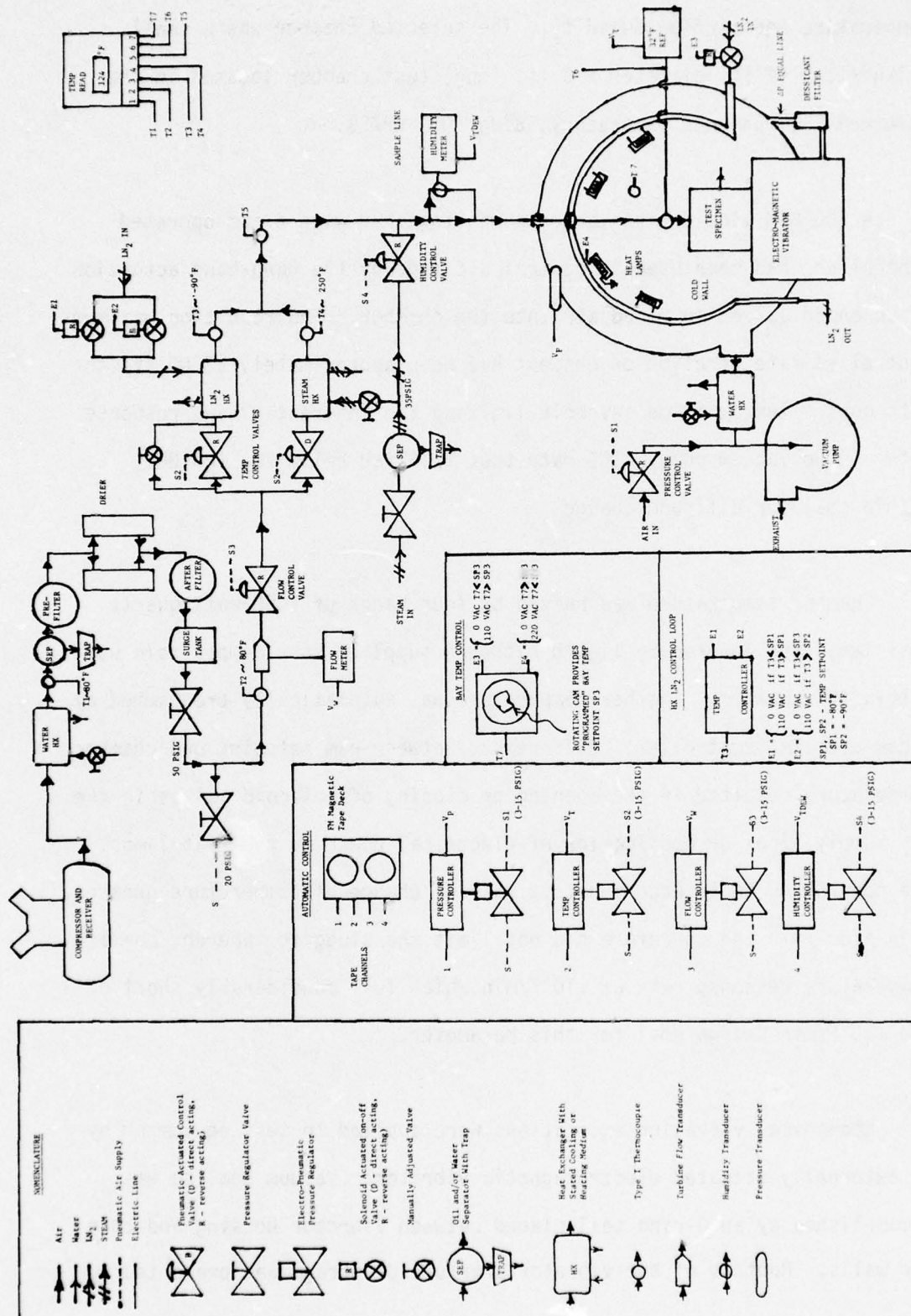
Figure 4. Schematic Diagram of Experimental CERT Facility

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temperature and airflow humidity. The selected chamber was a small, cylindrical (4 ft. diameter x 5 ft. long) test chamber located in the Combined Environments Laboratory, Bldg. 93, WPAFB.

A 300 CFM liquid ring vacuum pump, together with a cam-operated controller, had been used to control altitude by the bang-bang actuation of solenoid valves to bleed air into the chamber. The resulting maximum controlled rate of climb or descent had been approximately ±3000 ft/min with controller dynamics severely limiting the inherently high response rate of the vacuum pump. The rate thus fell far below the ±20,000 ft/min goal for altitude change.

Chamber temperature was raised by four banks of 1000 watt quartz heat lamps and lowered by liquid nitrogen supplied to a copper cold wall within the chamber. Chamber temperature was automatically programmed by a cam-operated controller. Differences between cam setpoint and chamber temperature resulted in the opening or closing of solenoid valves in the LN_2 supply lines or application of electrical power to the heat lamps. The controller could accommodate a rate of change of temperature greater than ±10°F/min and therefore did not limit the sluggish inherent chamber temperature response rate of ±10°F/min which fell considerably short of the ±30°F/min design goal for this parameter.

Mechanical vibration excitations were applied to test equipment by an externally situated electromagnetic vibrator. Vacuum sealing was accomplished by an O-ring seal placed between vibrator housing and chamber walls. Rupture of the vibrator armature diaphragm was prevented by

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a pressure equalization line connecting the chamber with the underside of the shaker table.

The electromagnetic vibrator permitted sinusoidal dwell or sweep and random vibration testing. Maximum force was 8000 lb; corresponding acceleration capability was dependent upon avionics weight. The desired vibration spectrum was achieved by manually adjusting 26 filters in the 5 to 2000 Hz frequency range. Overall amplitude of excitation could be manually raised or lowered as required.

As a matter of record, capabilities of the test chamber prior to modification were as follows:

Chamber Temperature: -100 to 300°F

                          ±10°F/min

Altitude: 0 to 66,500 ft

                          ±3000 ft/min controlled, zero through-flow

                          66,500 ft/min uncontrolled, zero

                                through-flow

Vibration: Sinusoidal or Random

                          5 to 2000 Hz

                          8000 lb maximum force

                          Manual setting of spectrum

                          Manually adjustable overall amplitude

Of these initial capabilities, only vibration met performance
specifications. Chamber static temperature response was considerably
less than needed for full flight profile simulation. Since static
temperature is a secondary environment, this compromise in performance
was accepted to facilitate timely and economical development of the
experimental facility. The inherent capability of the vacuum pump for
dynamic altitude simulation was compatible with performance goals.
However, modifications of the altitude control hardware had to be in-
stituted to meet the response rates required by control under through-
flow conditions.

The redesigned mode of chamber pressure (altitude) control consists
of connecting the liquid ring vacuum pump with the flow of continuously
controlled ambient air through a large pneumatic control valve directly
into the pump inlet. Total volumetric flow through the pump is thus
maintained constant while the chamber exhaust rate is varied for pres-
sure control. An alternative means of control would have consisted of
throttling pump flow. However, for this type of vacuum pump there is no
power saving resulting from the latter method. Greater pump wear would
inevitably result from frequent operating point changes as opposed to
the continuous steady-state operation characteristic of the chosen mode
of chamber pressure control. As a result chamber altitude can be kept
practically at sea level with the pump in operation.

It may be noted that the use of a liquid ring type of vacuum pump
is particularly appropriate for evacuating the continuous through-flow

of humidified air from the chamber since it is the only positive dis-
placement style of pump with wet air handling capability.

As an additional precaution, to protect the pump from extremely hot
or cold air emerging from the chamber, the exhausted air is tempered in
a water-cooled, shell-and-tube heat exchanger located in the line be-
tween pump and chamber.

3.3 PRESSURIZATION

Pressurization is accomplished by means of a 150 SCFM, laboratory
stock, reciprocating, positive displacement compressor to a pressure
between 85 and 100 psig.

3.4 CLEAN-UP

The clean-up process consists of removing liquid oil and water and
their vapors resulting from the compression process in order to prevent
subsequent fouling of the air-liquid N_2 heat exchanger in the thermal
conditioning section and for wide range humidity control in the humidi-
fication portion of the airflow loop. Equipment used to accomplish the
clean-up are an air-water heat exchanger, a liquid separator and trap,
an oil vapor filter and a twin tower, cycled, purge regenerated, des-
sicant air dryer. Dessicant dustings from the dryer are collected in an
after filter to prevent fouling of downstream components. A surge tank
and pressure regulating valve were included to isolate subsequent sec-
tions from the flow and pressure fluctuations caused by frequent cycling
(every 2-1/2 minutes) of the dryer and of the compressor at low process
airflow rates.

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3.5 FLOW CONTROL

The flow of process air is metered by pneumatically varying the flow area of a sliding gate type flow control valve. Volumetric airflow is measured by means of a turbine type flow meter located in the pressure controlled segment contained between the pressure regulator and flow control valve. Mass flow rate, W, is proportional to volumetric flow rate, Q, due to the following stable air properties existing in this segment of the loop:

Temperature, T = 80 ±5°F

Dewpoint Temperature = -90 ±5°F

Pressure, P = 50 ±1.5 psig

$$\frac{dW}{W} = \frac{dP}{P} + \frac{dT}{T} + \frac{dQ}{Q} = .04 + \frac{dQ}{Q} \tag{1}$$

3.6 THERMAL CONDITIONING OF PROCESS AIR

Temperature conditioning is achieved by dividing the total flow between parallel hot and cold heat exchangers and then remixing the separate air streams. Identical, oppositely acting, ganged pneumatic valves located upstream of the heat exchangers provide rapid, precise temperature control by distributing the proper amount of air to each heat exchanger. These valves, together with the flow control valve, are located in the benign environment preceding the heat exchangers to circumvent a requirement for special high or low temperature capability. The cooling and heating fluids for temperature conditioning the streams of air to be mixed were chosen from the standpoint of their availability, suitability for the temperature range to be covered, compatibility with existing services and installations, and low usage cost. These were the liquid nitrogen already in use for static chamber temperature

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control and medium pressure plant stream, economical because of its large-scale production and the need for it in air humidification.

Both heat exchangers were of the shell-and-tube type with conditioning media supplied to the shell side and air flowing through the tube bundles. The heat exchangers were oriented vertically with steam supplied to the upper shell-side port and liquid nitrogen to the lower shell-side port of their respective exchangers in order to aid the separation of liquid and vapor phases. Significant savings in procurement cost and time were effected by the purchase of a standard size tube and shell type of heat exchanger constructed of 316 stainless steel for use as the liquid nitrogen/air exchanger instead of a higher priced, custom order cryogenic heat exchanger.

The air/LN$_2$ heat exchanger was provided with a feedback control loop to circumvent unstable temperature fluctuation conditions. This was accomplished by controlling LN$_2$ flow to the heat exchanger so as to maintain the LN$_2$ heat exchanger air outlet temperature approximately constant. The feedback loop depends upon a signal of instantaneous temperature of air leaving the heat exchanger. Accordingly, the small airflow through this heat exchanger, required at all times in order to prevent a reading of stagnant air temperature, was provided by a manually-controlled bypass valve placed around the temperature control valve. The bypass flow causes a reduction in maximum temperature of airflow obtainable since a small amount of cold air is always mixed with the hot airstream.

3.7 HUMIDIFICATION

Immediately preceding the chamber, a controlled amount of steam from a 35 psig supply is injected into the airstream through a choked gate valve. Humidity is controlled by variation of the valve orifice area and monitored by a dewpoint sensor located in a line that samples the airstream, cools it to protect the sensor, and then returns it at a downstream point so that total flow is not affected. The principle of humidity transducer operation is based on the change in dielectric constant due to polarization of water vapor molecules. The sensor is accurate with a fast response. Nevertheless, some sacrifice in response rate is entailed by use of the sampling loop. Moreover, accurate measurement of saturated air with entrained moisture is not possible. This is not a problem since environments are kept slightly below saturation in the simulation. While saturated environments are uncommon during flight, aircraft ECS are being designed to prevent this condition.

A minor complication was introduced by use of a dewpoint sensor to control specific humidity whereas dewpoint is a function both of specific humidity and pressure (altitude). This problem was overcome by precomputation of a single setpoint command signal for dewpoint based on the desired instantaneous values of both flow specific humidity and chamber pressure (altitude). This approach proved practical because chamber pressure responds much more rapidly than does humidity and doubtlessly contributed to control stability. This approach also obviated the need to include analog computational elements in the control loop with input from a pressure transducer.

3.8 CONTROLLABILITY ASPECTS

It will be noticed that the described system exhibits a large measure of decoupling of the controlled parameters and their means of control. Operation of the flow control valve at pressure ratios greater than critical (choked flow) renders the flow rate essentially independent of chamber pressure. The mixing approach to airflow temperature control guarantees a degree of independence from variations in flow rate. Decoupling of parameter control modes enhances system stability. The reverse is also true to a considerable extent. Changes in flow rate will affect chamber altitude; hence, an adjustment of the altitude control valve is required to maintain a constant chamber pressure in the face of changing flow rates.

3.9 DESIGN FOR MANUAL CONTROL

The capability for continuous manual control of the facility demanded by the first phase of development was provided pneumatically through four, hand-operated pressure regulators. Each regulator was mounted on a control console in front of a seated human operator. The four regulators provided a 3-15 psig pneumatic signal to the valve controlling the respective environmental parameter: airflow rate, airflow temperature, airflow humidity and altitude. It was determined beforehand that the desired rates of change of the environments lay within the response capabilities of human operators. The control loop is completed by means of two, two-pen X-Y recorders on which the desired time trace of each parameter sensor signal was pre-plotted. The human operator can observe the instantaneous deviation of the sensor trace from the desired one and make the necessary regulator adjustment.

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3.10 DESIGN FOR AUTOMATIC CONTROL

Continuous setpoint control of altitude, airflow temperature, airflow rate, and airflow humidity is accomplished by magnetic tape control. Continuous 1 to 5 VDC analog signals are played from four tracks of a magnetic tape and input to four electronic controllers. All four environmental parameter transducer outputs are also converted to a 1 to 5 VDC signal and input to the controllers. The resulting functional relationships between environmental parameter and controller input voltage are given below.

Environmental Parameter	Controller Input Voltage

P - Altitude
(Pressure - psia)
$$V_P = \frac{4P}{15} + 1 \tag{2}$$

T - Airflow
Temperature °F
$$V_T = \frac{4000}{6} \left(a(T-32) + b(T-32)^2 \right) + \frac{14}{6} \tag{3}$$

W - Airflow Rate
(lb/min)
$$V_W = \frac{4W}{6.37} + 1 \tag{4}$$

TDEW - Airflow Humidity
(Dewpoint
Temperature - °F)
$$V_{TDEW} = \frac{4}{10} f(TDEW) + 1 \tag{5}$$

where $a = 0.021$ mv/°F, $b = 0.0000141$ mv/(°F)^2, and $f(TDEW) = 0.675$ exp $(0.02185\ TDEW)$

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The above equations are used to pre-program the magnetic tape with the desired environmental profiles in order to provide continuous set-point input to the controllers. Each controller compares tape input with transducer input by means of a control equation which uses three control functions: gain, integral and derivative. For heuristic purposes, the control equation that governs the controller output voltage E may be expressed as:

$$E = K_1(PV - SP) + K_2 \int (PV - SP) \, dt + K_3 \frac{d}{dt} (PV - SP) \qquad (6)$$

where PV = process variable (1 to 5 VDC modulated transducer input) and SP = setpoint (1 to 5 VDC tape input). The "K" factors are the function (gain, integral and derivative) weighting factors which can be adjusted independently to obtain optimum control.

Each controller output voltage, E, is input to an electropneumatic pressure regulator which provides a corresponding 3 to 15 psig pneumatic signal to the appropriate environmental parameter control valve. This pneumatic signal results in a readjustment of control valve position, thereby changing transducer signal. This process is continuous and induces a rapid convergence toward balanced tape and transducer controller input signals. The schematic diagram depicting a typical control loop is shown in Figure 5.

The actual magnetic control tape programming can be most easily and accurately handled by a digital to analog computer program. This approach uses sufficient data points (time and parameter) to define the

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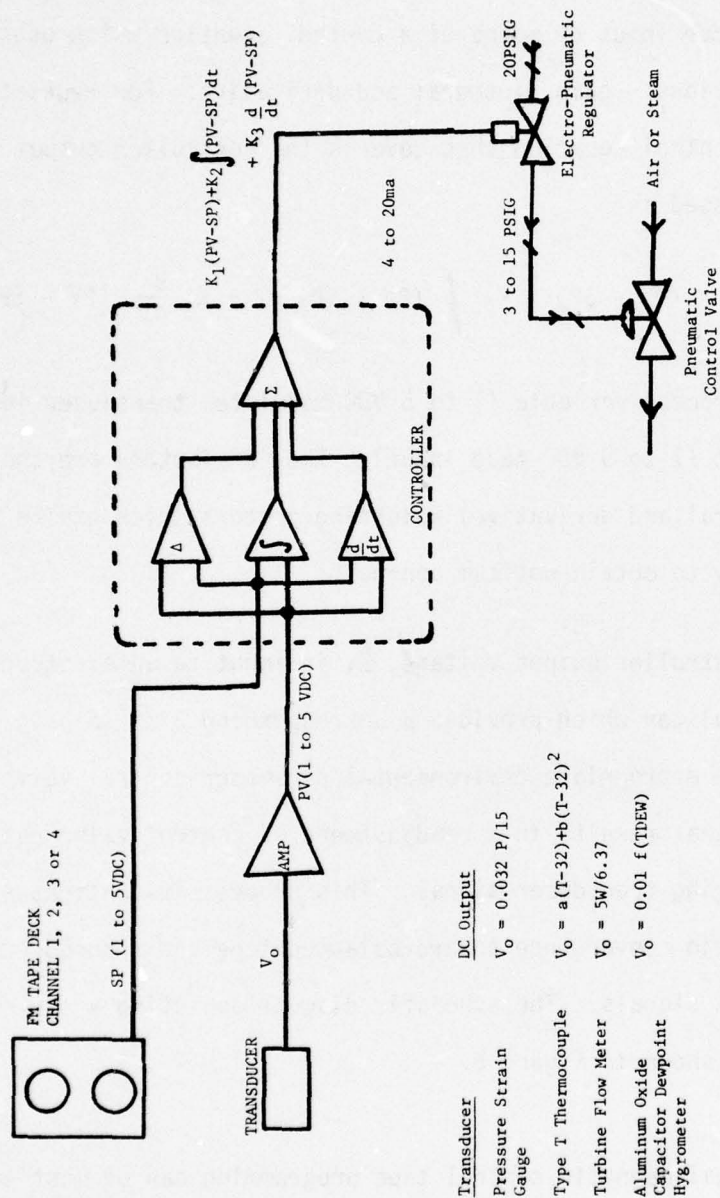
Figure 5. Schematic Diagram for Magnetic Tape Control of CERT Facility

Transducer	DC Output
Pressure Strain Gauge	$V_o = 0.032\ P/15$
Type T Thermocouple	$V_o = a(T-32)+b(T-32)^2$
Turbine Flow Meter	$V_o = 5W/6.37$
Aluminum Oxide Capacitor Dewpoint Hygrometer	$V_o = 0.01\ f(TDEW)$

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environmental profile, extrapolates linearly with time between data points, and converts physical parameter values to equivalent DC voltages. However, most hybrid computers have FM recording systems with standard ±1.0 VDC levels at ±40% deviation from center frequency rather than the 1 to 5 VDC range used in this application. The appropriate equations for tape preparation on a standard FM recorder are easily obtained by subtracting 3 from V_P, V_T, V_W, and V_{TDEW} and dividing them by 2.

Since typical environmental profiles are four hours in duration, a rather slow playback tape drive speed is required. The FM recording system used in this application has 15/16-inch per second tape drive speed and 10-inch tape reels, thereby providing up to 13 hours continuous setpoint control.

4. COMPONENT SIZING

The construction of a system for dynamic simulation of altitude and forced air environments requires careful sizing of all components to ensure that the resulting system performance meets design goals. General considerations in sizing major system components are given below.

4.1 COMPRESSOR AND AIR DRYER

The compressor size is principally determined by the maximum flow rate required to condition projected test items. Additional compressor flow capacity must be available for pneumatic controls, air dryer regeneration and line losses. A heatless, regenerative type dessicant dryer,

as used in the experimental facility, provided low installation and op-
erating costs. However, the airflow pressure transients associated with
frequent dryer cycling and the large percent (up to 33%) of total flow
required for regeneration posed problems in facility development. Their
solution is described in paragraph 3.4. The primary reason for select-
ing the heatless type dryer, as opposed to an alternative long cycle,
heat reactivation dryer, was low development cost. The use of a 150
SCFM laboratory stock compressor and the selected dryer presented the
most economical choice for experimental development. However, it is
emphasized that careful matching of compressor and dryer is generally
required to minimize system costs because of the manner in which purge
flow requirements and air pressure affect dryer performance.

4.2 VACUUM PUMP

The introduction of forced air into the test chamber strongly im-
pacts its altitude (pressure) capability. This effect is analyzed in
Appendix A and yields a chamber pressure response assuming constant
pumping rate, Q_{OUT}, and mass inflow, W_{IN}, described by

$$P = P_o \frac{T}{T_o} \left[\frac{W_{IN}}{\rho_o Q_{OUT}} + \left(1 - \frac{W_{IN}}{\rho_o Q_{OUT}}\right) e\left(-t \frac{Q_{OUT}}{V}\right) \right] \qquad (7)$$

where P_o = atmospheric pressure (14.7 psia), t = time (min), V = chamber
volume (60 ft^3) and P = chamber pressure. Note that maximum altitude is
governed by the ratio of forced airflow rate to pump flow rate while
altitude response rate is governed by both this ratio and the ratio of
pump flow rate to chamber volume (easily seen by differentiating the
above expression).

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The facility vacuum pump has a volumetric Q_{OUT} capacity of approximately 300 CFM while maximum forced airflow rate is 6 lb/min. Therefore

$$\text{Time Constant} = \frac{V}{Q_{OUT}} = 0.2 \text{ minutes}$$

$$\begin{array}{c}\text{Min Pressure} \\ \text{at Max Flow}\end{array} = \frac{P_o T}{T_o} \frac{W_{IN}}{\rho_o Q_{OUT}} = 3.8 \text{ psia (33,000 ft)}$$

This analysis shows that the existing vacuum pump would not meet performance specifications for maximum altitude at maximum flow. However, as shown in Table I, altitude change rates appear satisfactory over a broad flow range up to 90% of maximum altitude in each case. This compromise in design was accepted to reduce development costs of this feasibility model.

TABLE I

THEORETICAL MAXIMUM ALTITUDE AND RATE
OF CLIMB CAPABILITIES OF CERT I CHAMBER

At .9 H_F

W_{IN} (lbm/min)	P_F (psfa)	H_F (kft)	H (kft)	P (psfa)	$\dot{P}$ (psf/min)	dP/dH (psf/ft)	$\dot{H}$ (kft/min)
1	92.	70.	60	151.	294	.0074	39.4
2	185.	55.	50	242.	289	.012	24.2
4	369.	41.	37	453.	418	.022	19.2
6	553.	33.	30	629.	376	.029	13.1

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4.3 HEAT EXCHANGERS

The sizing of heat exchangers is a straightforward process once the conditioning media (steam, LN_2, water, etc), maximum flow and desired outlet temperatures have been selected. As previously discussed, the use of an air-to-LN_2 heat exchanger for cold air conditioning requires automatic control of the heat exchanger outlet temperature. This problem occurs because full airflow at minimum outlet temperature sizes the heat exchanger. Therefore, at low flow conditions in this heat exchanger outlet air tends to approach LN_2 temperature (-320°F) with resulting flow and temperature instability.

4.4 VALVES

Sizing of valves will not be considered beyond pointing out that flow, and inlet and outlet pressure dictate the required valve flow coefficient C_v. Extreme examples of sizing pertain to the humidity and altitude control valves.

The humidity valve is a steam service valve; maximum steam flow is calculated as follows:

Max Steam Flow = Max Humidity Ratio x Max Flow

$$= 0.031 \; \frac{lb \; H_2O}{lb \; air} \times 6 \; \frac{lb \; air}{min} = 0.19 \; lb \; \frac{H_2O}{min}$$

This corresponds to a flow of only approximately 11 lb steam/hour. Thus for a 35 psig steam a very small C_v (<0.21) is required.

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Sizing of the altitude control valve is governed by the zero altitude (valve full open) condition. Maximum flow through the altitude control valve is the vacuum pump flow rate (approximately 300 CFM = 23 lb/min) minus the conditioned flow W injected into the chamber. This flow creates a pressure drop ΔP across the valve so that chamber pressure is less than ambient atmospheric pressure at the test facility location. This is effectively a nonzero altitude condition. However, this problem can be minimized by selecting the proper valve size. The pressure drop across the valve is related to flow by the following formula recommended by the Fluid Controls Institute (Reference 13)

$$\frac{W}{\rho_{STD}} = 1360 \; C_v \sqrt{\frac{\Delta P}{T}} \sqrt{\frac{P_1 + P_2}{2}} \tag{8}$$

where ρ, T and P are the ambient atmospheric conditions at the test facility. Therefore, selecting 0.3 psia as the maximum allowable pressure drop results in the following requirement for the altitude control valve flow coefficient.

$$C_v = \frac{60(23)}{1360(0.076) \sqrt{0.3}} \sqrt{\frac{530}{14.7}} = 190 \tag{9}$$

For nonzero process airflow the pressure drop ΔP can be calculated as follows:

$$\Delta P = 0.3 \left(\frac{23 - W}{23} \right)^2 \tag{10}$$

This calculation illustrates how the minimum altitude capability improves with increased through flow.

It will be noticed that sizing of the altitude control valve also governs the descent rate from altitude conditions. An estimate of the time for descent from maximum altitude is calculated in Appendix A where it is found that the time to descend to 90% of sea level pressure is 2 seconds.

4.5 DUCTING

The choice of material, length and diameter is not critical for most ducting in the forced air simulator. The only basic requirements are adequate size to pass maximum flow without excess pressure drop and sufficient strength to avoid buckling under maximum pressure differential across the ducting. An exception is the section of ducting between the hot and cold airstream mixing point and test item inlet. The length, size and type of ducting material in this section is critical to the airflow temperature rate of change because air in the ducting exchanges heat with the duct which in turn is heated or cooled by room air and chamber air. The ducting material should have a low heat capacity per unit length. This ensures that thermal equilibrium occurs in minimum time. These considerations suggest that a good, economical choice for this ducting is small diameter, thin wall copper tubing. In practice, a short section of flexible ducting must be used, connecting the copper tubing to the equipment inlet in order to permit equipment vibration testing.

The principal effects of duct length and material on airflow temperature response are evident by considering the duct to be adiabatically insulated from chamber and room air so that only duct and airstream exchange heat. With this assumption, transient response of air

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flowing through a duct is analyzed in Appendix B. In particular, an expression is derived for the inlet air temperature variation required to maintain a constant rate of change of exit temperature over a given period of time or temperature range. The evaluation of this expression is programmed for a digital computer in Appendix C. Table II gives results of sample calculations in which duct diameter, length, and airflow rate are varied with and without the presence of a 16 lb valve in the line. Standard copper tubing is considered.

The conclusions derived from these calculations are that duct diameter and length should be kept small, while valves in the line should be avoided if possible.

The airflow temperature response is directly proportional to specific heat and length of ducting and is inversely proportional to airflow rate. Consequently, obtaining rapid temperature change becomes increasingly difficult at low airflow rates. Maintaining a high airflow rate through the ducting and proportioning the required amount to the avionics test item appears at first sight to be a plausible approach. The problem, however, still lies with practical implementation, which would result in considerable complications to the overall control design. Moreover, an unacceptably high amount of the response gains could be lost in the "proportioning device" or valve as is evident from the sample calculations of Table II.

TABLE II

DELIVERY DUCT AIR TEMPERATURE RANGE TO MAINTAIN
100°F/MIN RATE OF CHANGE BETWEEN -65°F AND 200°F

Flow (lbm/min)	Diameter (inches)	Length (ft)	Temperature (°F Difference)	Valve Mass (lb)
1.	.331	5.	320.	16.
1.	.444	5.	333.	16.
1.	.555	5.	351.	16.
1.	.680	5.	363.	16.
1.	.805	5.	374.	16.
1.	.214	3.	307.	16.
1.	.331	3.	312.	16.
1.	.444	3.	320.	16.
1.	.555	3.	330.	16.
1.	.214	3.	270.	0.
1.	.331	3.	275.	0.
1.	.444	3.	282.	0.
1.	.555	3.	292.	0.
1.	.680	3.	298.	0.

Note: Standard Copper Tubing

SECTION III

FACILITY PERFORMANCE

1. BASIC PERFORMANCE CAPABILITIES

The design concept selected for simulation of the forced-air-cooled airborne equipment environment, together with appropriate sizing and selection of system components, was found to meet essentially the stipulated performance specifications over the necessary parameter ranges. Use of an available laboratory stock compressor and retention of the existing chamber temperature control provisions, however, forced some compromise in the design goals for airflow and compartment temperature. A reduction in total airflow, and hence in total cooling capacity of the facility from that desired, resulted from the combined requirements of simulation, regeneration and control airflow rates exceeding the capacity of the chosen compressor. Use of the existing chamber temperature control entailed a relaxation of requirements for response speed of this secondary environmental parameter.

The experimentally determined ranges of the environmental parameters are given below:

Environment	Desired Range	Actual Range
Airflow Rate:	0 to 6 lb/min	0 to 5.73 lb/min
Process Air Temperature:	-65°F to 120°F	-90°F to 200°F
Process Air Humidity: (dewpoint temperature)	-65°F to 90°F	-90°F to 140°F
Altitude (at maximum flow):	0 to 60,000 ft	0 to 38,000 ft
Compartment Temperature:	-65°F to 250°F	-100°F to 300°F

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The completed facility has an inherent capability for extremely fast flow and humidity response, since both flow and humidity (rate of moisture injected) respond immediately to changes in their respective control valve orifice areas. Limitations on flow and humidity response are therefore principally determined by transducer and valve actuation response times. These response times are given below:

Parameter	Response Time
Airflow Rate:	Less than 2 seconds for full range change
Airflow Humidity:	5 seconds for 63% change in maximum moisture content

The effects of airflow rate and chamber temperature on airflow temperature response were experimentally determined by applying step changes in temperature control valve position while recording the response at various flow rates and chamber temperatures. The system was "tuned" prior to these experiments by setting the LN_2 flow control to maintain a cold heat exchanger outlet air temperature of approximately -90°F. The manual bypass valve was adjusted for a 180°F airstream mix temperature at maximum flow rate. The results of these experiments are shown in Figures 6 through 9.

It will be noticed that airflow temperature response is primarily governed by airflow rate. This effect may be roughly expressed by:

$$T(t) = T(0) + [T(\infty) - T(0)] [1 - \exp(- t/\tau)] \tag{11}$$

where t = time (sec), $T(t)$ = airflow temperature (°F) at time t, and τ = time constant (sec). For this experiment, the initial and final states were 160°F and -90°F for a temperature reduction and -60°F and
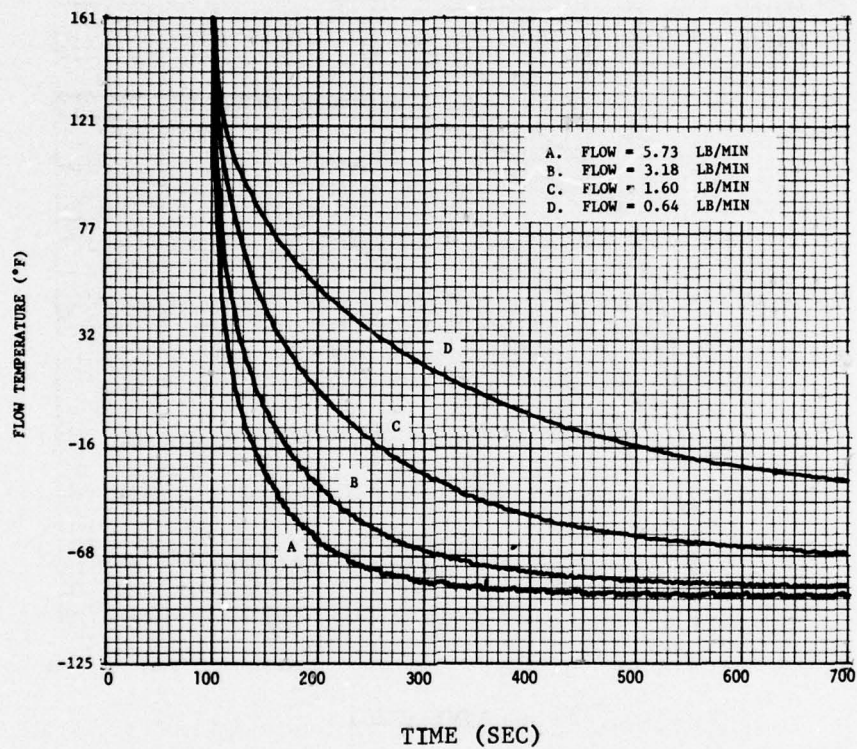
34

Figure 6. Temperature Response of Cooling Air to Maximum
Step Decrease in Temperature Setpoint at Various
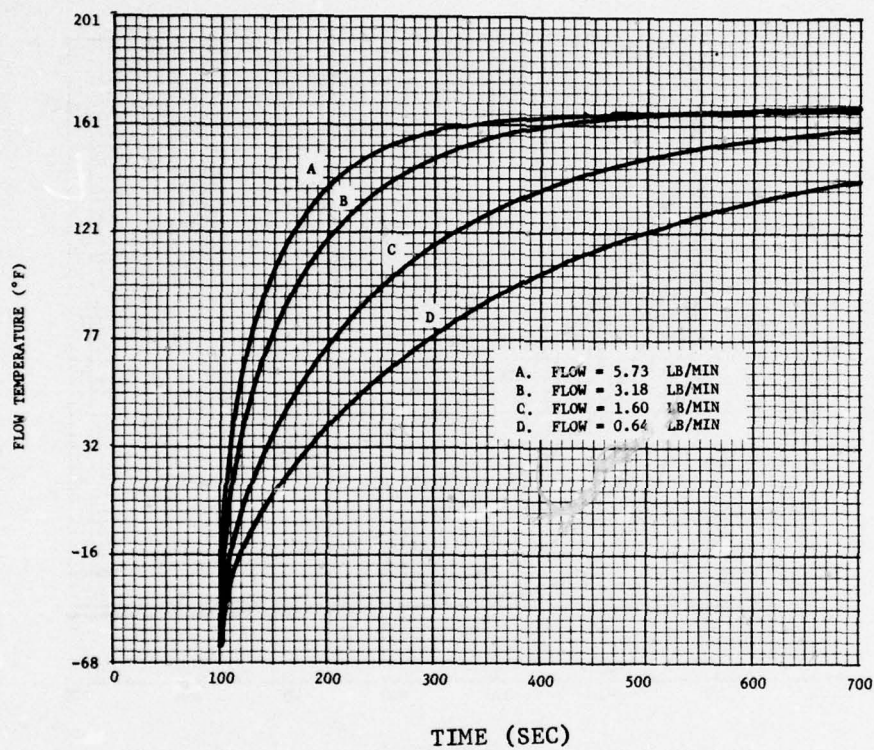Flow Rates (Bay Temperature = 70°F)

Figure 7. Temperature Response of Cooling Air to Maximum
Step Increase in Temperature Setpoint at Various
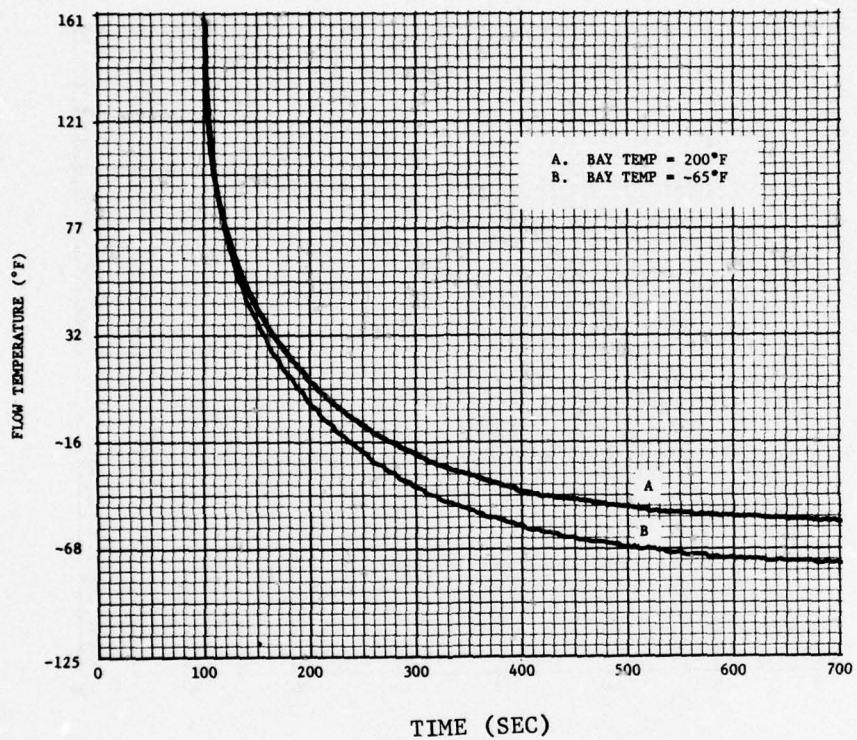Flow Rates (Bay Temperature = 70°F)

Figure 8. Effect of Bay Temperature on Airflow Temperature
Response to Maximum Step Decrease in Setpoint
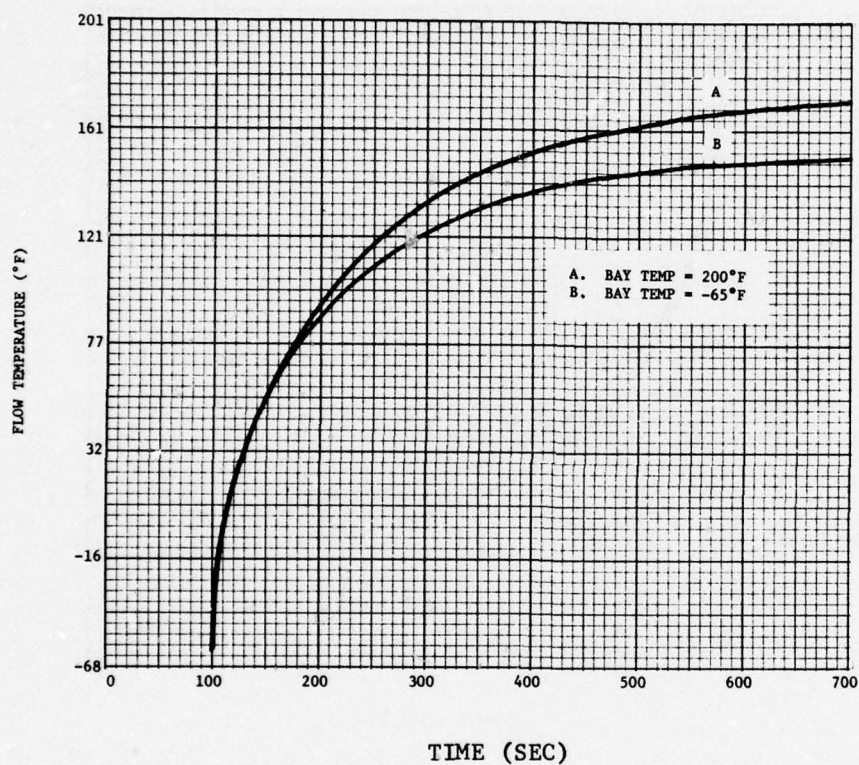(Airflow Rate = 1.6 lb/min)

Figure 9. Effect of Bay Temperature on Airflow Temperature
Response to Maximum Step Increase in Setpoint
(Airflow Rate = 1.6 lb/min)

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180°F for a temperature increase. A regression of the experimental results with the above expression produced the following inverse correlation of time constant, τ, in seconds with airflow rate, W, in lb/min for a 70°F chamber temperature:

$$\tau = 25 \, \frac{W_{max}}{W} = 25 \, \frac{(5.73)}{W} \tag{12}$$

This equation, it is seen, agrees with the analytical result of equation B-4 of Appendix B.

Altitude response range and rate were determined experimentally for three different airflow rates (0, 3.18 and 5.73 lb/min) as shown in Figure 10. Maximum altitude is strongly dependent on airflow rate and the results obtained are listed below.

Airflow Rate (lb/min)	Max Altitude (ft)
0	66,500
3.18	44,000
5.73	38,000

These results compare favorably with the theoretical values of Table I. Altitude response rate is also flow dependent, since time to reach maximum altitude at a given flow is a constant 60 seconds with a uniform time constant of approximately 12 seconds. Time to descend from maximum altitude is less than 10 seconds, but somewhat greater than the 2 seconds calculated in Appendix A due to neglect of pipe flow resistance. A convenient method of placing these results in perspective is to compare chamber altitude response rate with the new rate of climb records recently established by the F-15 aircraft. This aircraft established a
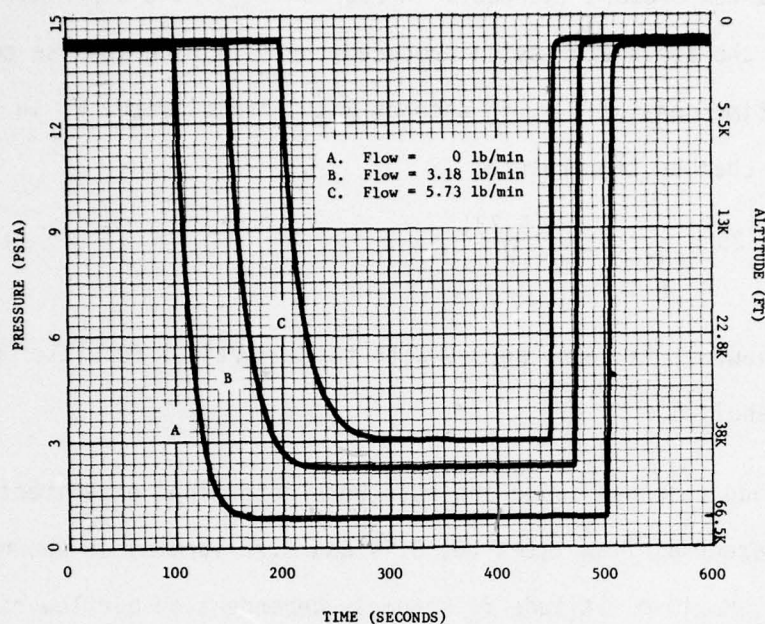
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Figure 10. Chamber Pressure (Altitude) Response to Maximum
Step Change in Setpoint at Various Flow Rates

rate of climb to 12,000 meters (39,370 ft) in 59.4 seconds. The test
chamber capability exceeds this record except at the highest flow con-
ditions.

2.   PERFORMANCE ON MANUAL CONTROL

Continuous manual control of airflow humidity was found to be ex-
tremely difficult. A typical run without humidity control as made by
three human operators is shown in Figures 11 and 12. The degree to
which the desired traces are followed is considered quite satisfactory.
Approximately seventy hours of reliability testing of units of an air-
borne radar system were performed using the manual mode of facility con-
trol prior to arrival and installation of the automatic control equip-
ment. Figures 11 and 12 depict examples of portions of the environmental
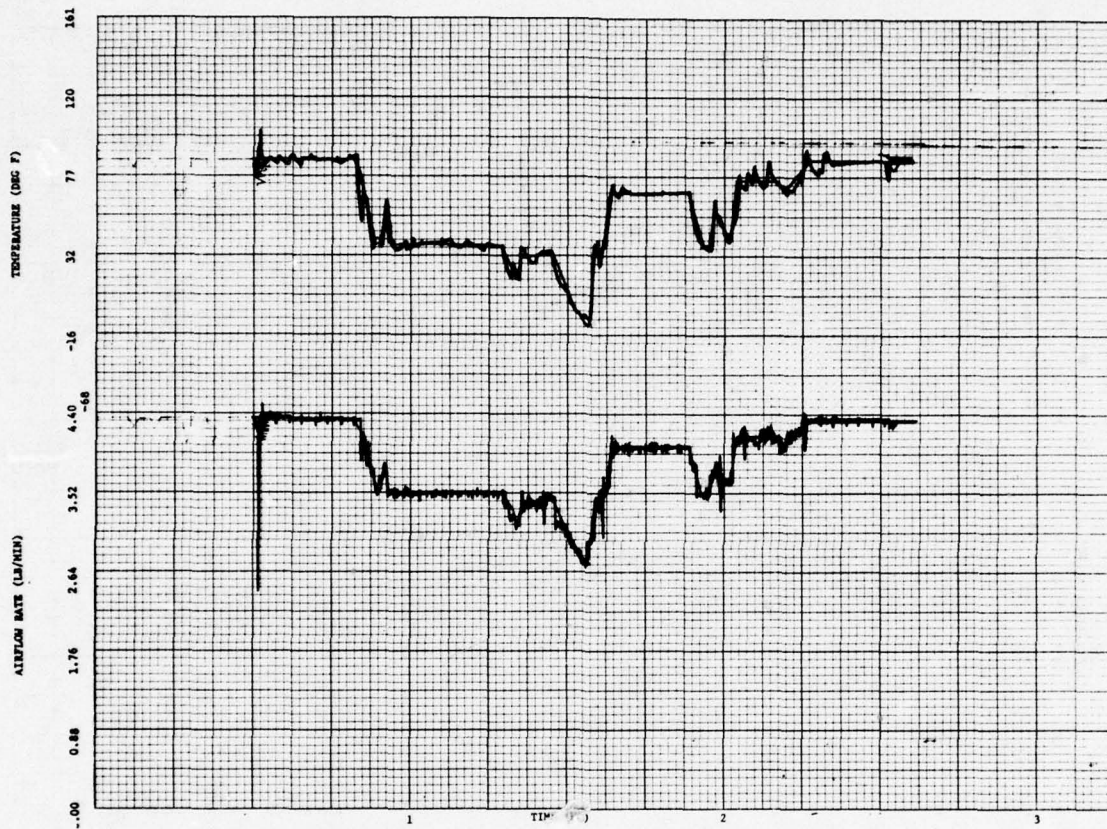profiles used in this CERT test.

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Figure 11. Manual Control of Airflow Rate and
Airflow Temperature Profiles

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Figure 12. Manual Control of a Pressure (Altitude) Profile

3. PERFORMANCE WITH AUTOMATIC CONTROL

Checkout of automatic control first required tuning the level and mix of the three controller modes for each controlled parameter. This was accomplished by subjecting each parameter to repeated step changes in setpoint and adjusting the corresponding three mode control settings to obtain damped parameter response.

A magnetic control tape was manually programmed to provide a series of ramp inputs which covered the full parameter ranges. The three values of times for full range change were 200 seconds, 100 seconds and 50 seconds. This control tape was used to check out single parameter, automatic control performance. During this performance check, the tape input and parameter transducer output were plotted simultaneously on a two-pen X-Y plotter. Pen separation on the two-pen plotter corresponds to 15 seconds. Hence, perfect control corresponds to two identical curves with a 15-second separation.

The experimental results for single parameter automatic control are shown in Figures 13 through 18 for airflow, pressure, humidity and temperature, respectively. Inspection of these results illustrates that input control tape signals covered the design goal ranges. Excellent parameter control was obtained within the system's capability. Oscillations were avoided at all three control rates (200, 100 and 50 seconds for full range). This result verifies that proper control mode adjustments had been used to obtain both an adequate rate of response and an ensured stable response.
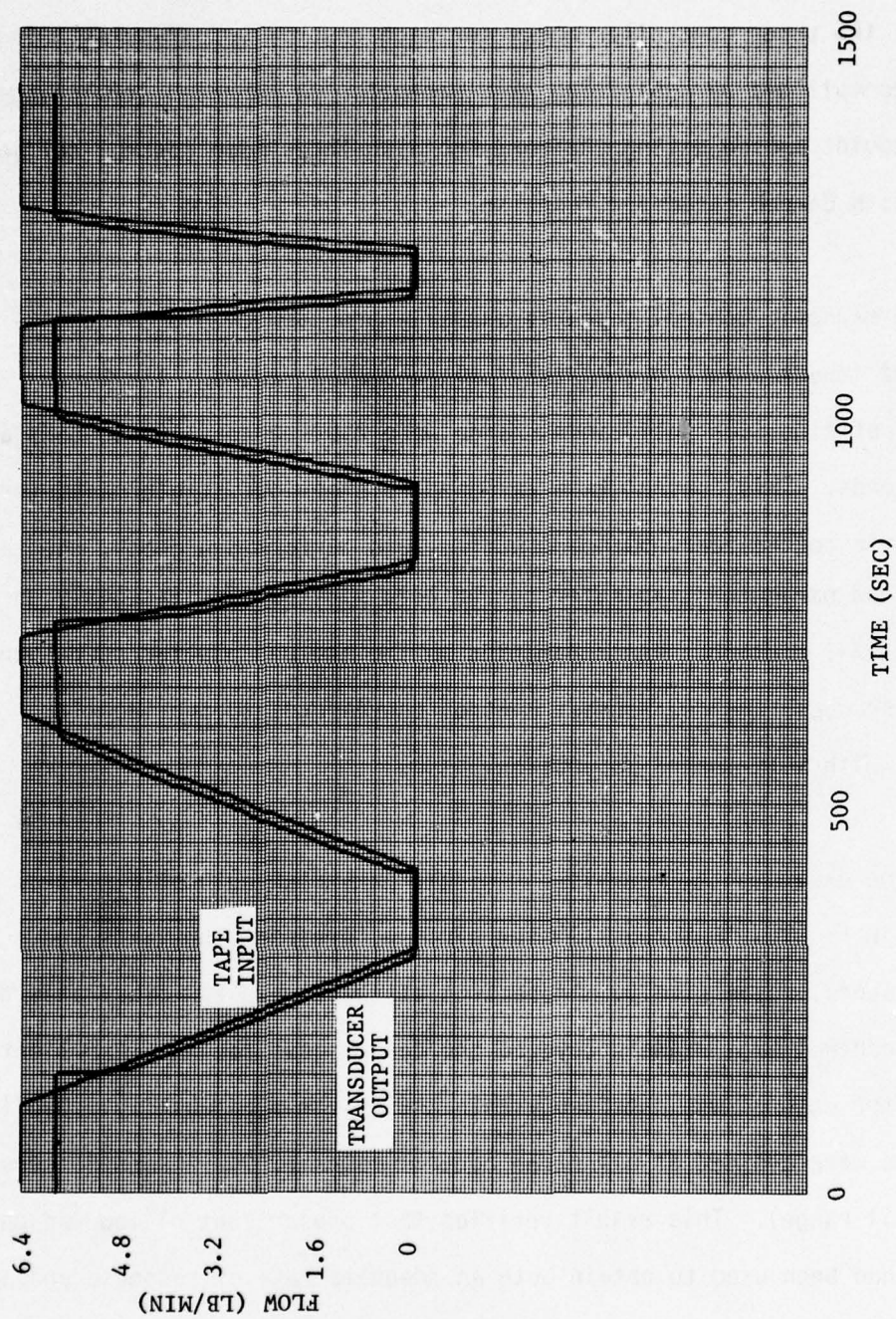
Figure 13. Airflow Rate Response to Automatically Controlled Ramp Changes
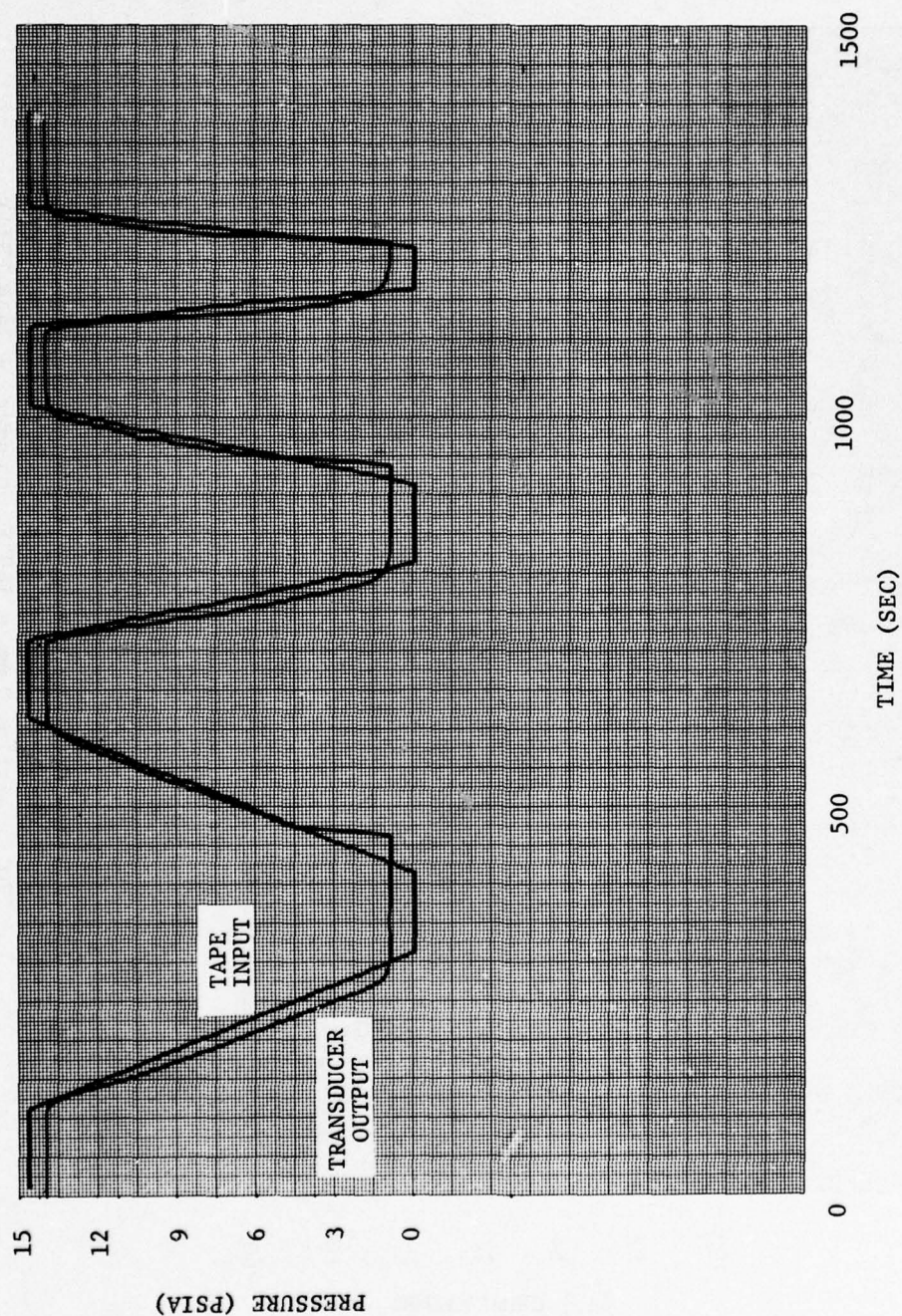
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Figure 14. Pressure (Altitude) Response to Automatically Controlled Ramp Changes (Airflow Rate = 0 lb/min)

Figure 15. Airflow Humidity Response to Automatically Controlled
Ramp Changes (Airflow Rate = 1.6 lb/min, Bay Temp =
70°F, Airflow Temp = 140°F, Pressure = 14.4 PSIA)
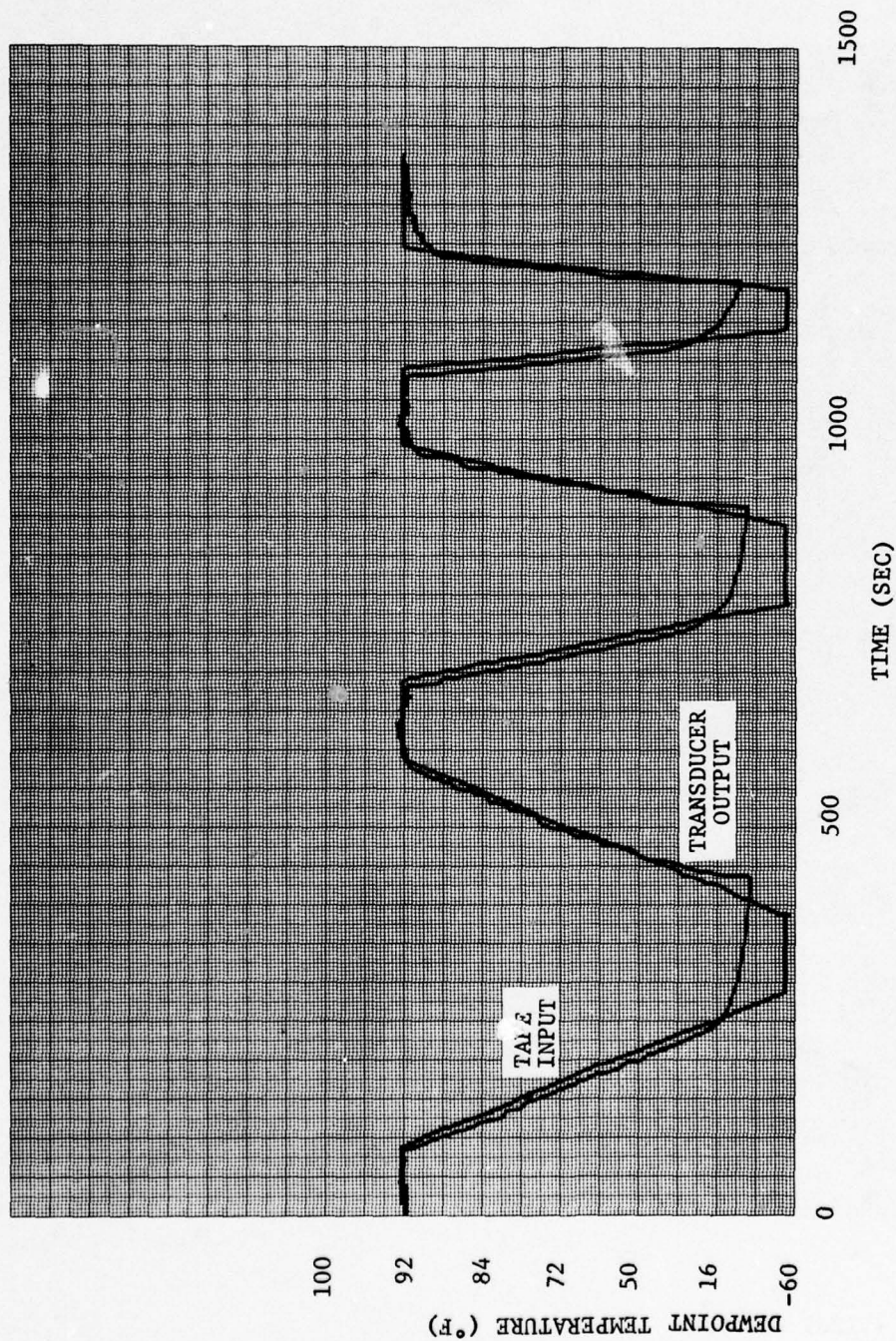
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Figure 16. Airflow Humidity Response to Automatically Controlled
Ramp Changes (Airflow Rate = 5.7 lb/min, Bay Temp =
70°F, Airflow Temp = 140°F, Pressure = 14.4 PSIA)

Figure 17. Airflow Temperature Response to Automatically Controlled
Ramp Changes (Airflow Rate = 1.6 lb/min, Airflow Dewpoint =
-70°F, Bay Temp = 70°F, Pressure = 14.4 PSIA)

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Figure 18. Airflow Temperature Response to Automatically Controlled Ramp Changes (Airflow Rate = 5.7 lb/min, Airflow Dewpoint = -70°F, Bay Temp = 70°F, Pressure = 14.4 PSIA)

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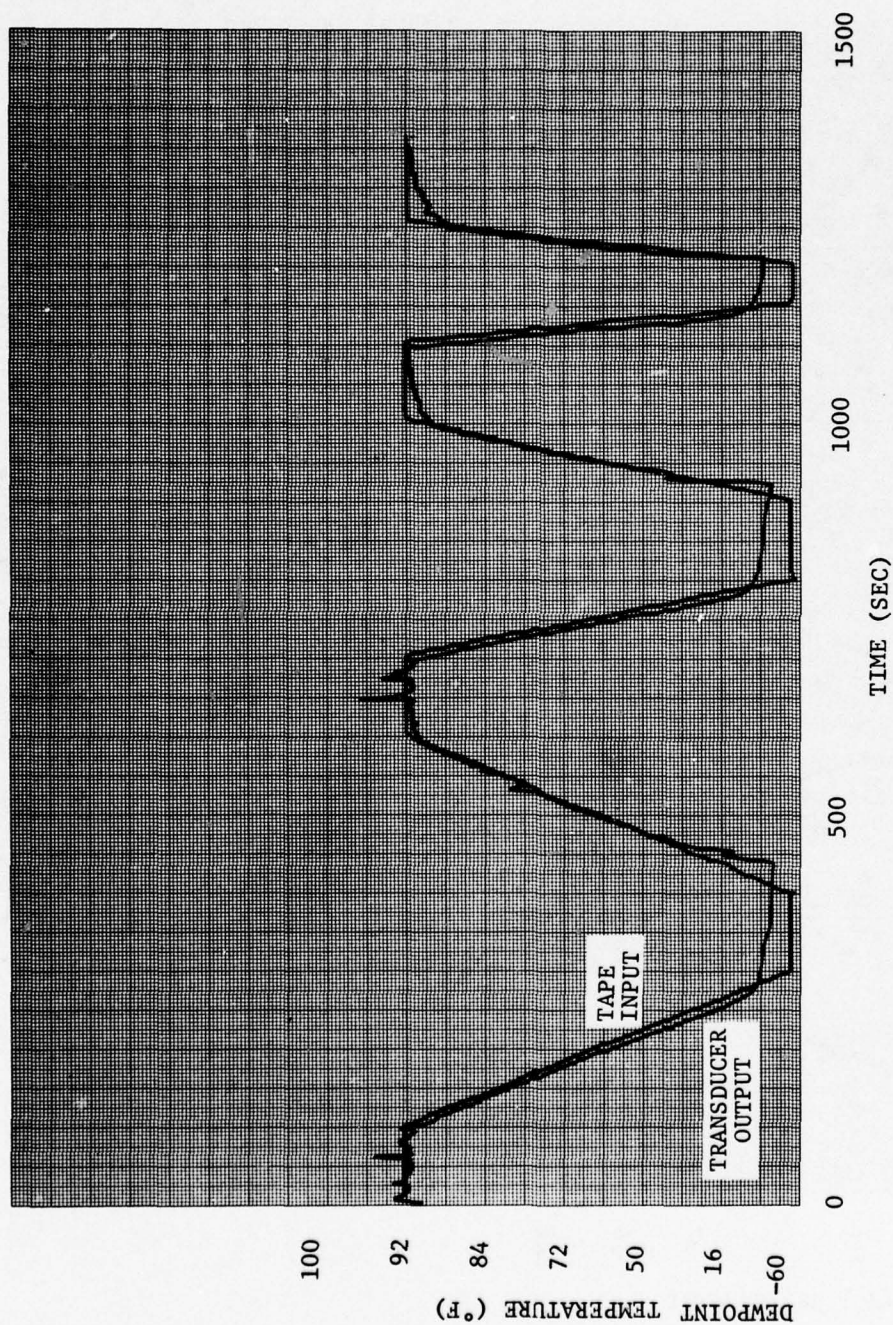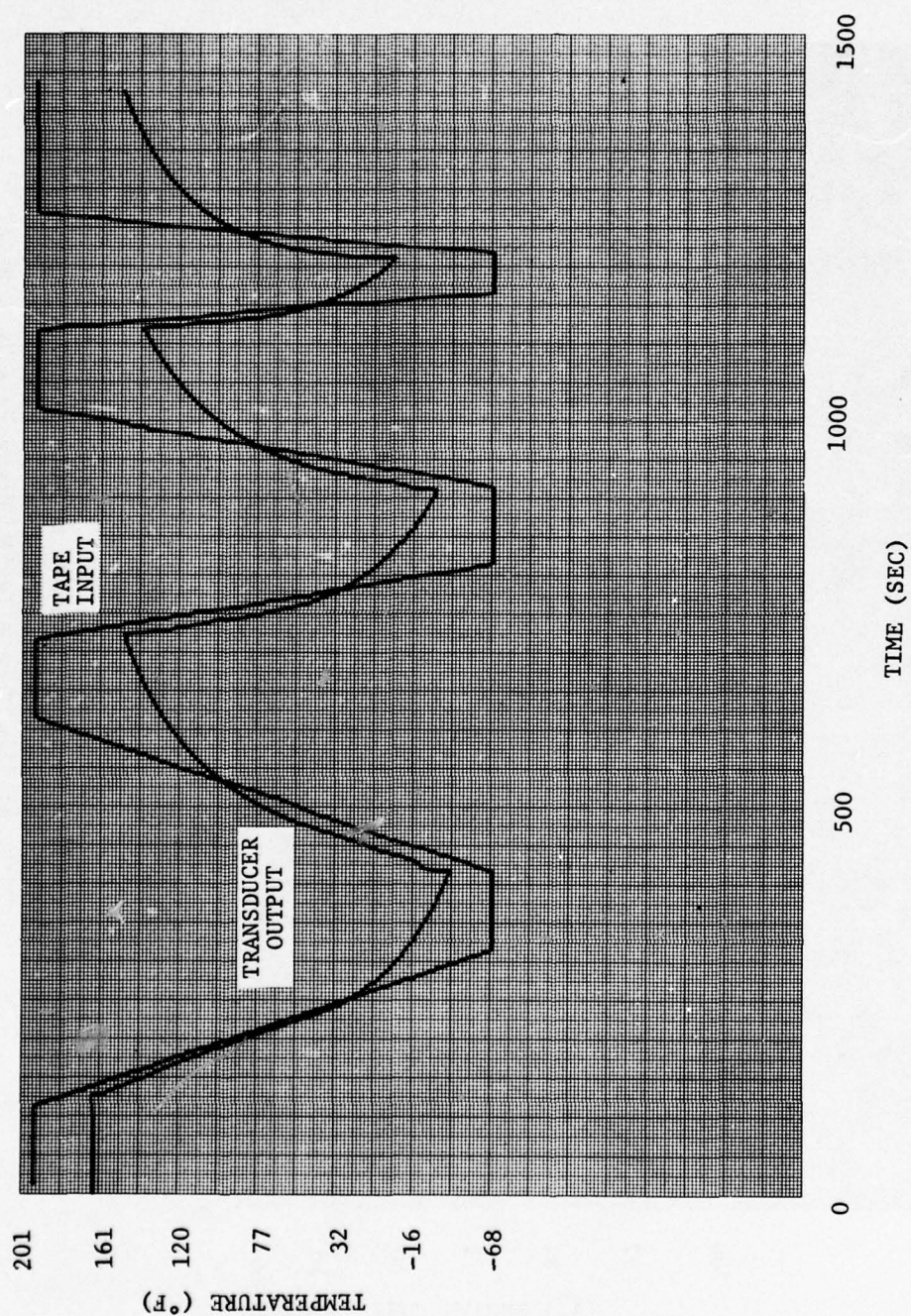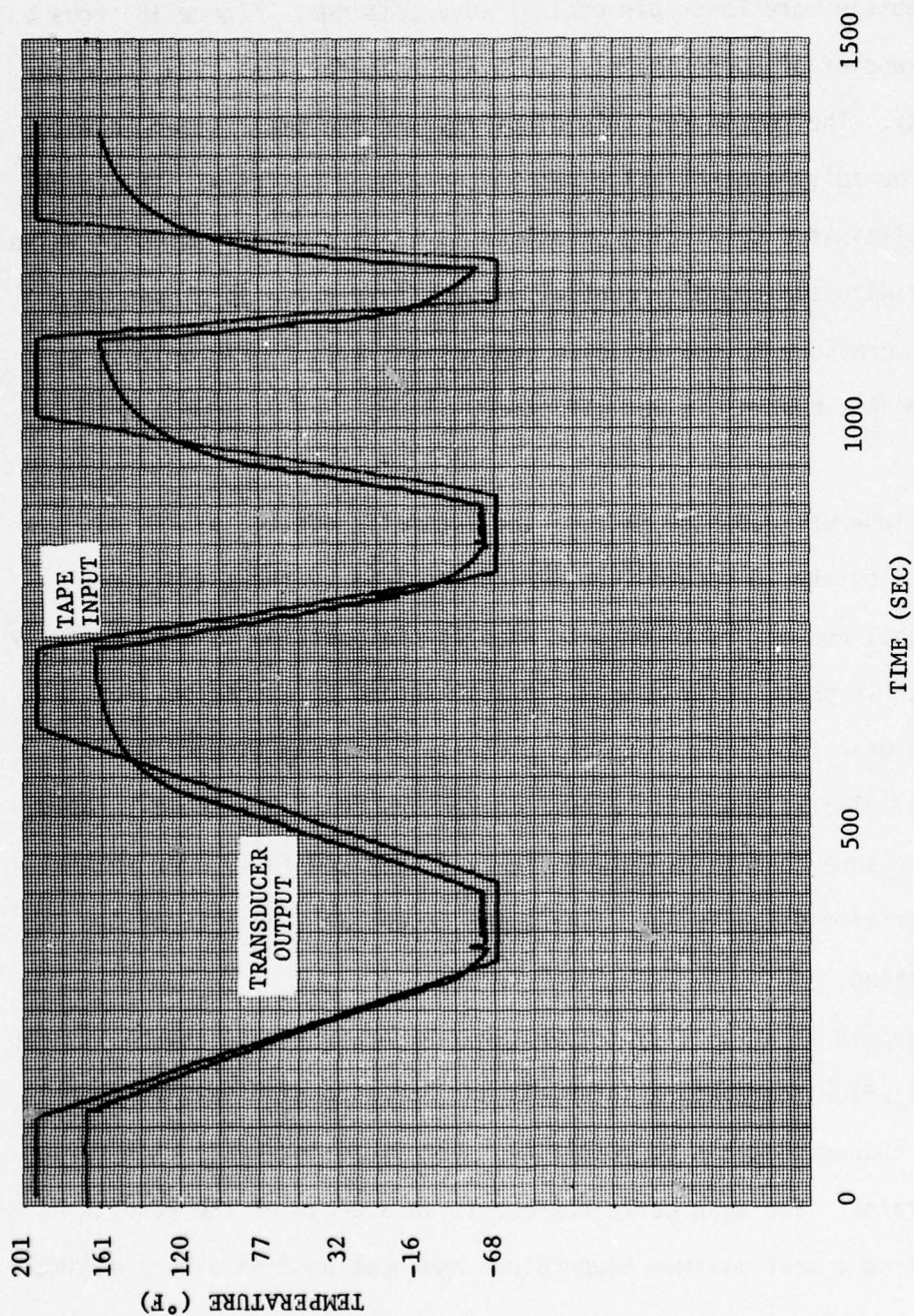
Humidity (dewpoint temperature) was found to be the most difficult parameter to automatically control even though repeated efforts were made to obtain more favorable control mode settings. Figure 15 shows a lowest dewpoint of approximately -20°F which is far less than the dryer capability. The reason for this disparity was the small steam leakage when the humidity control valve was in the closed position. This problem was eliminated by placing an extremely tight shut-off solenoid valve in series with the humidity control valve. This valve is closed when pneumatic pressure to the humidity control valve is 3 psig or less so that automatic control is not affected.

The inherent response rate of the automatic control system is determined by the 10-second time period required by the electropneumatic signal convertors to actuate when initial setting is 3 psig (fully closed) or 15 psig (fully open). This 10-second delay can be seen in Figure 13 when tape input requires a change from zero flow or maximum flow. The electro-pneumatic signal convertors respond extremely rapidly to all pressure conditions between fully closed and fully open so that controller time delay is not significant except at endpoints of the parameter range.

Figure 19 is a facility temperature/temperature rate capability map showing airflow temperature intervals and corresponding maximum ramp rates of change that can be maintained over these intervals with various airflow rates. The mean curve represents an average of the results of cooling from a near maximum temperature and heating from a near minimum temperature. These extremes are shown as the limits of variance bands about the curves.

Figure 19. Airflow Temperature Interval Versus Rate of Change
for Various Airflow Rates
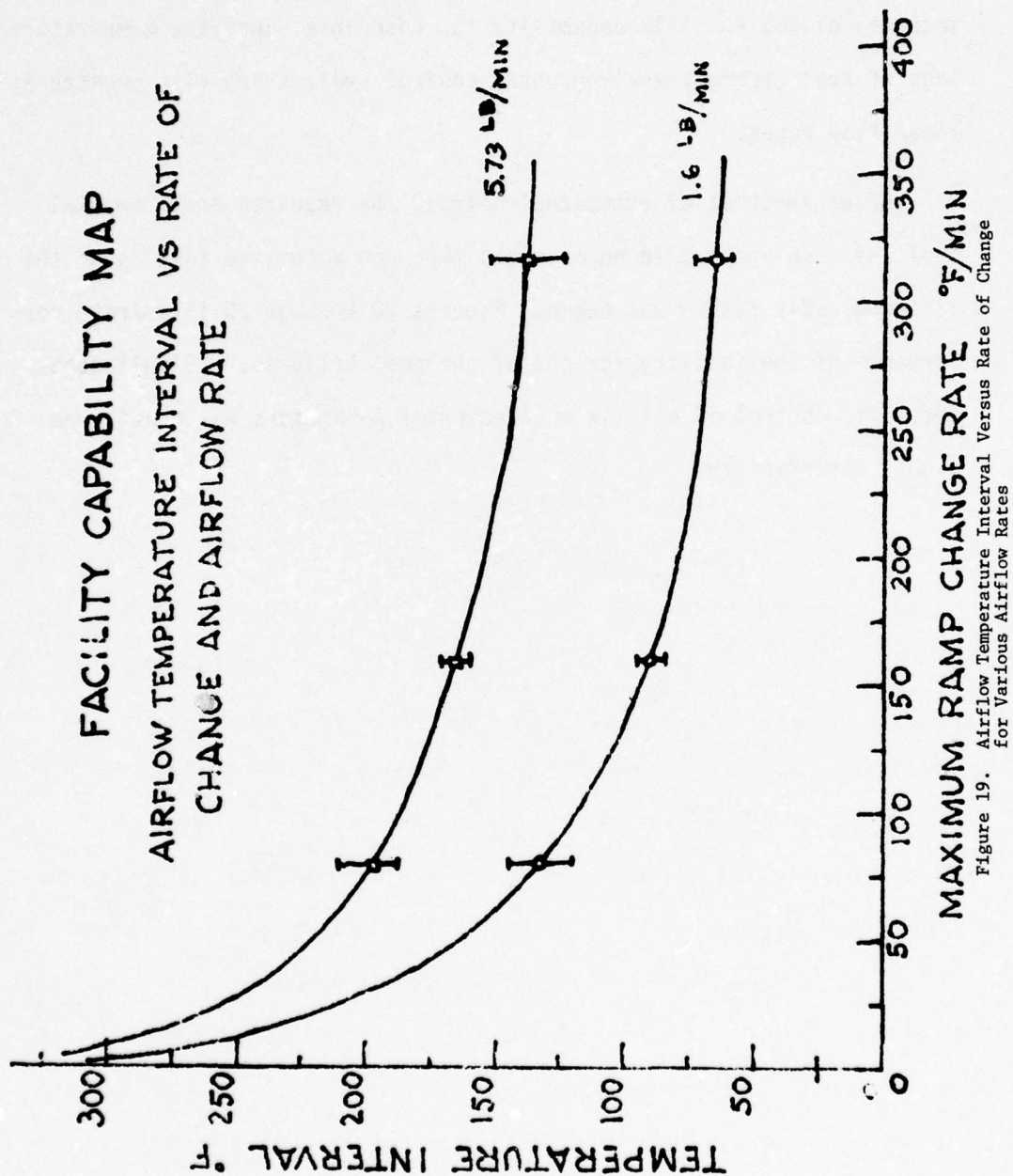
It is evident from this map that the air temperature change rates of 30°F/min and 80°F/min can be achieved at the minimum and maximum flows of 1.6 and 5.7 lb/min, respectively, over the required temperature interval of 185°F. This capability is reasonable since the temperature lags of real aircraft environmental control systems are also greater at lower flow rates.

After checkout of automated control, the required environmental profiles were programmed on magnetic tape and automated testing of the airborne radar system was begun. Figures 20 through 25 illustrate performance of the facility for one of the test "flights." Simultaneous automatic control of all six environmental parameters was considered highly satisfactory.
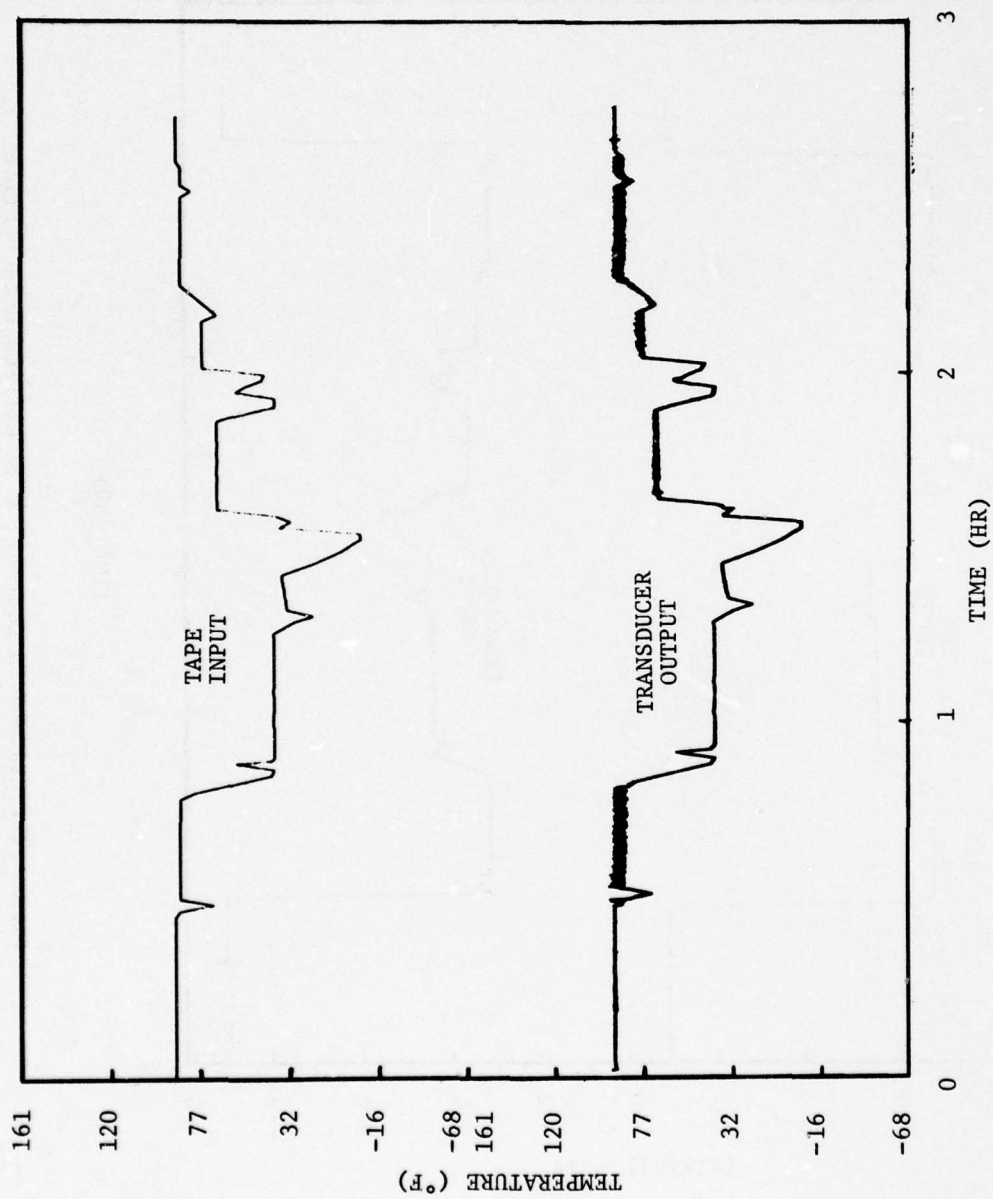
Figure 20. Automatic Control of an Airflow Temperature Profile
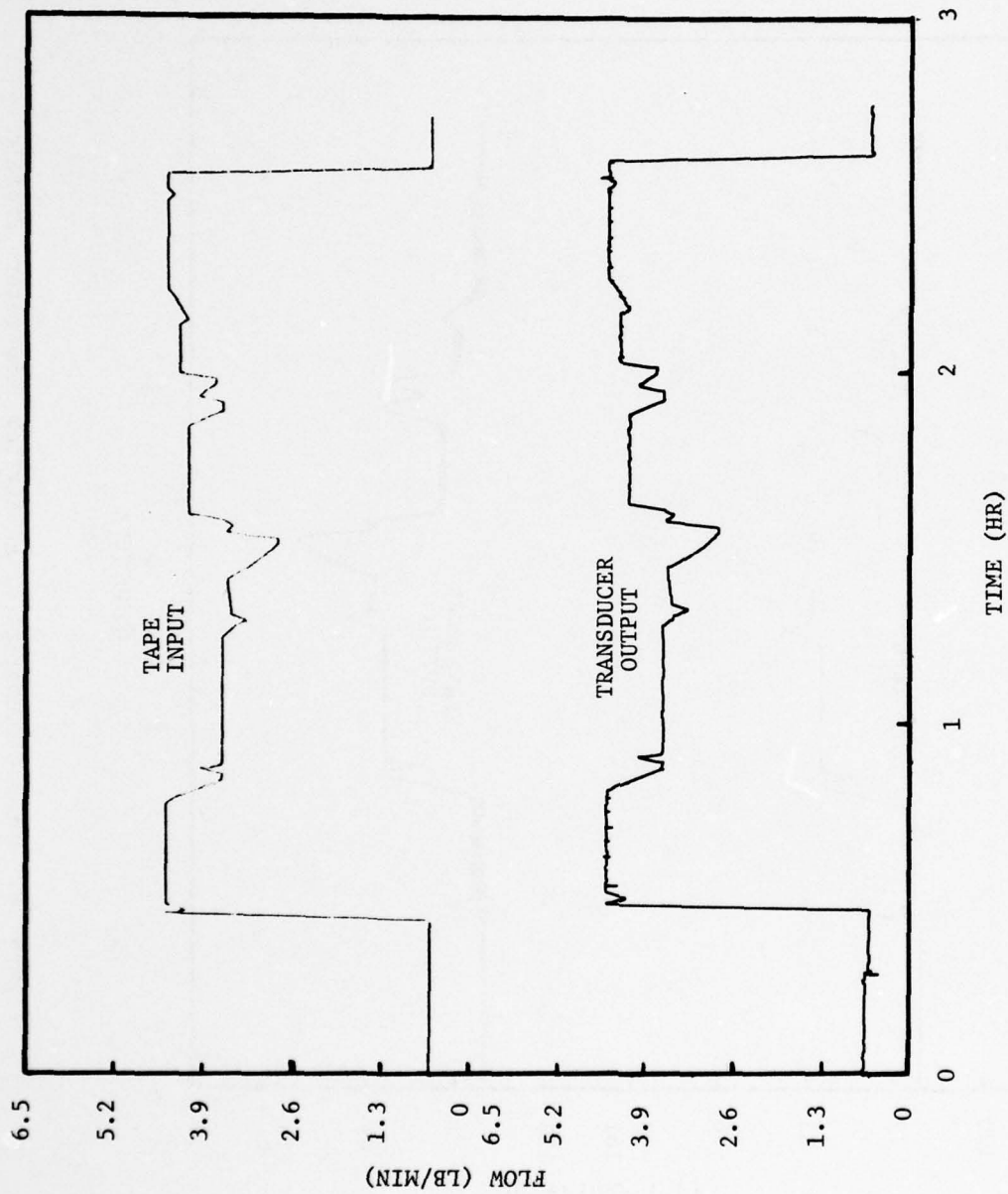
53

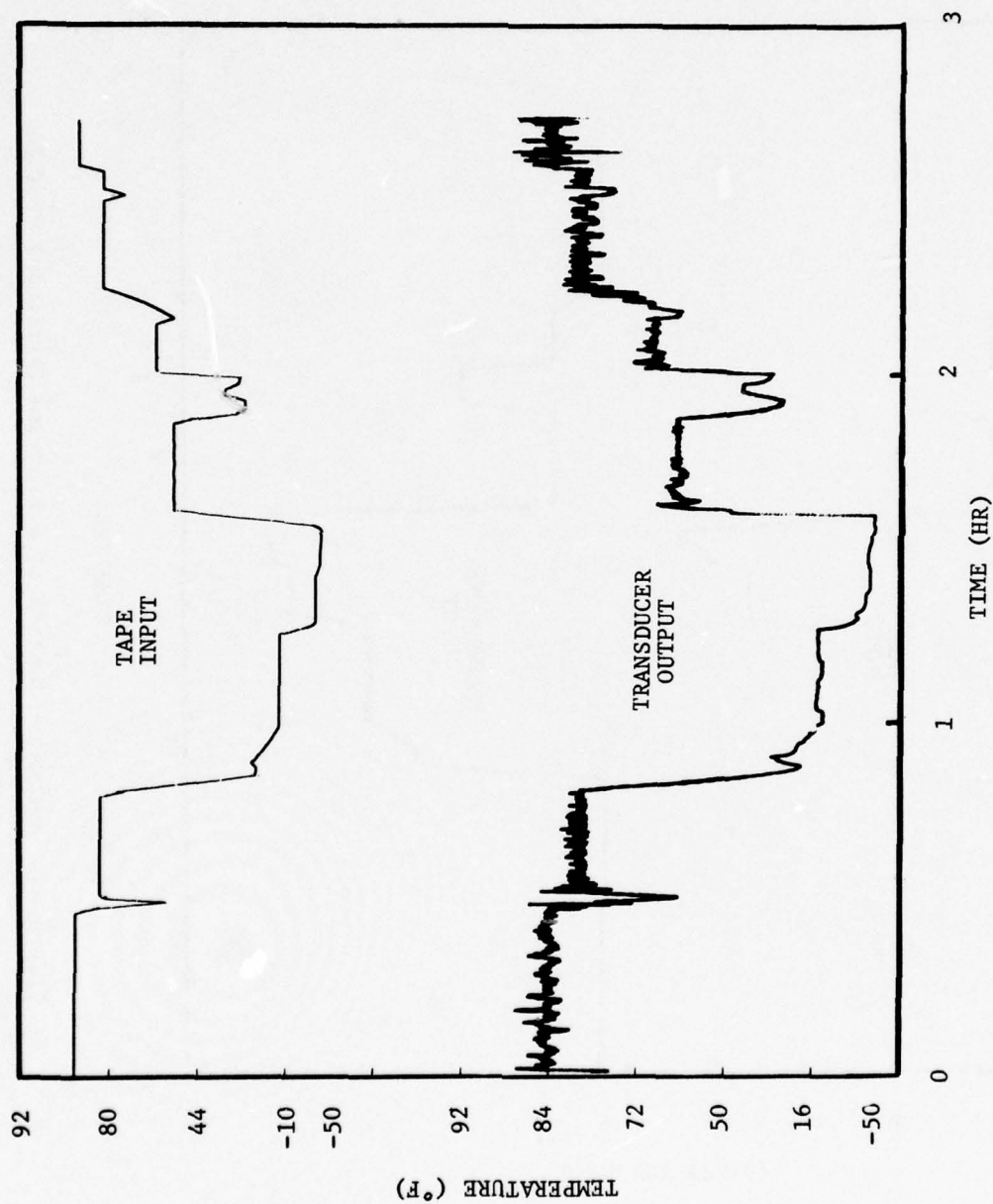Figure 21. Automatic Control of an Airflow Rate Profile

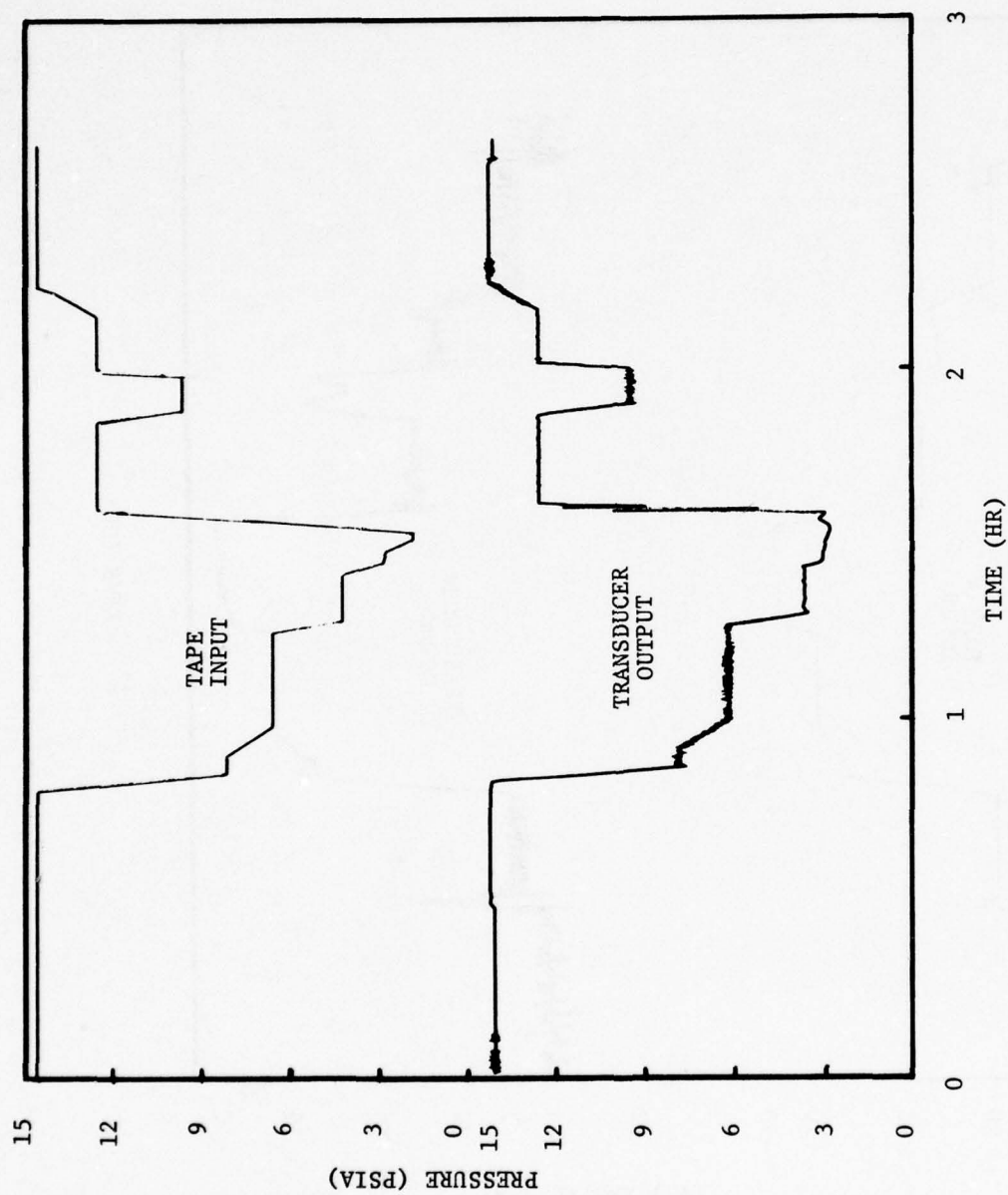Figure 22. Automatic Control of an Airflow Humidity Profile

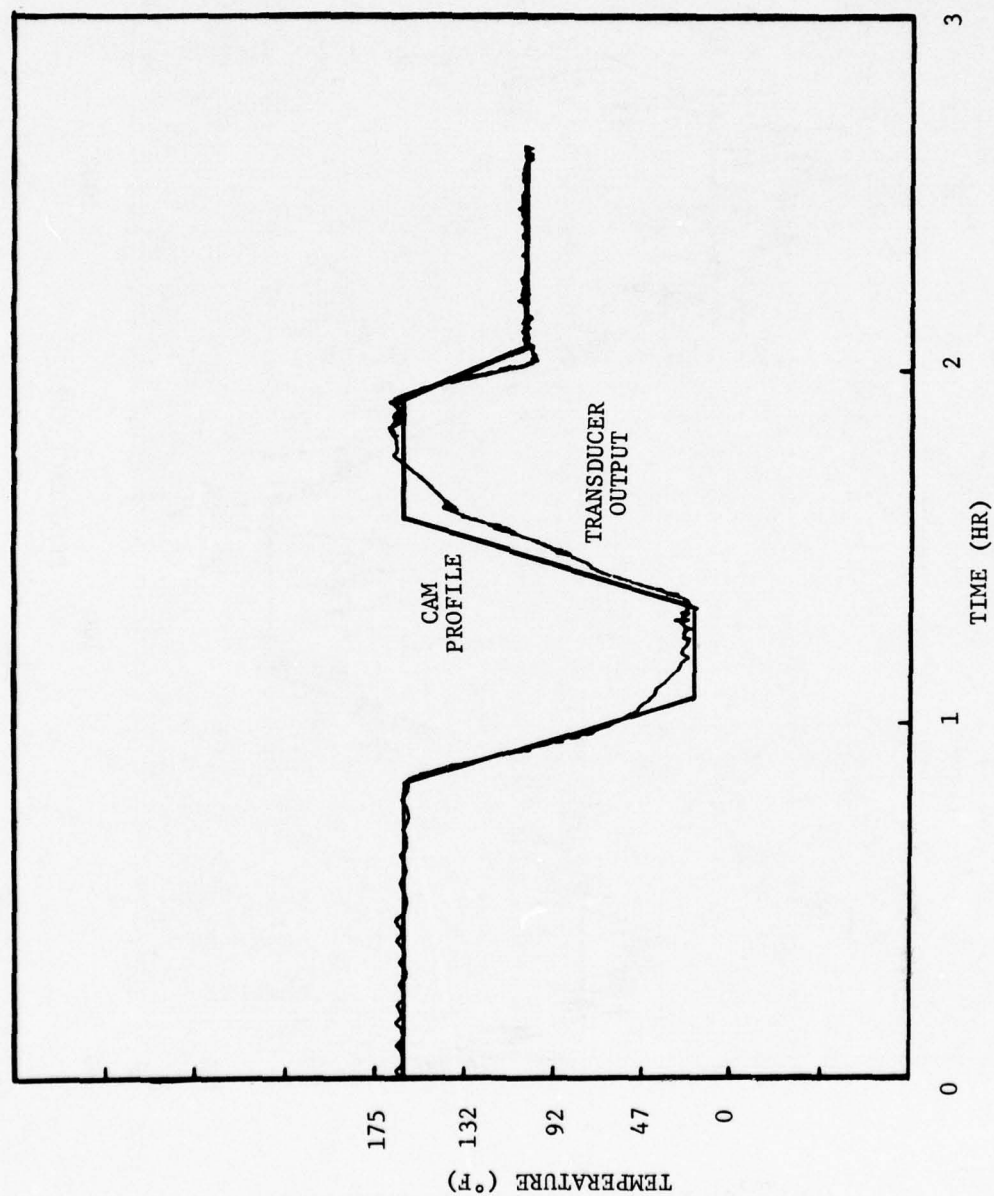Figure 23. Automatic Control of a Pressure (Altitude) Profile

Figure 24. Automatic Control of a Chamber (Bay) Temperature Profile
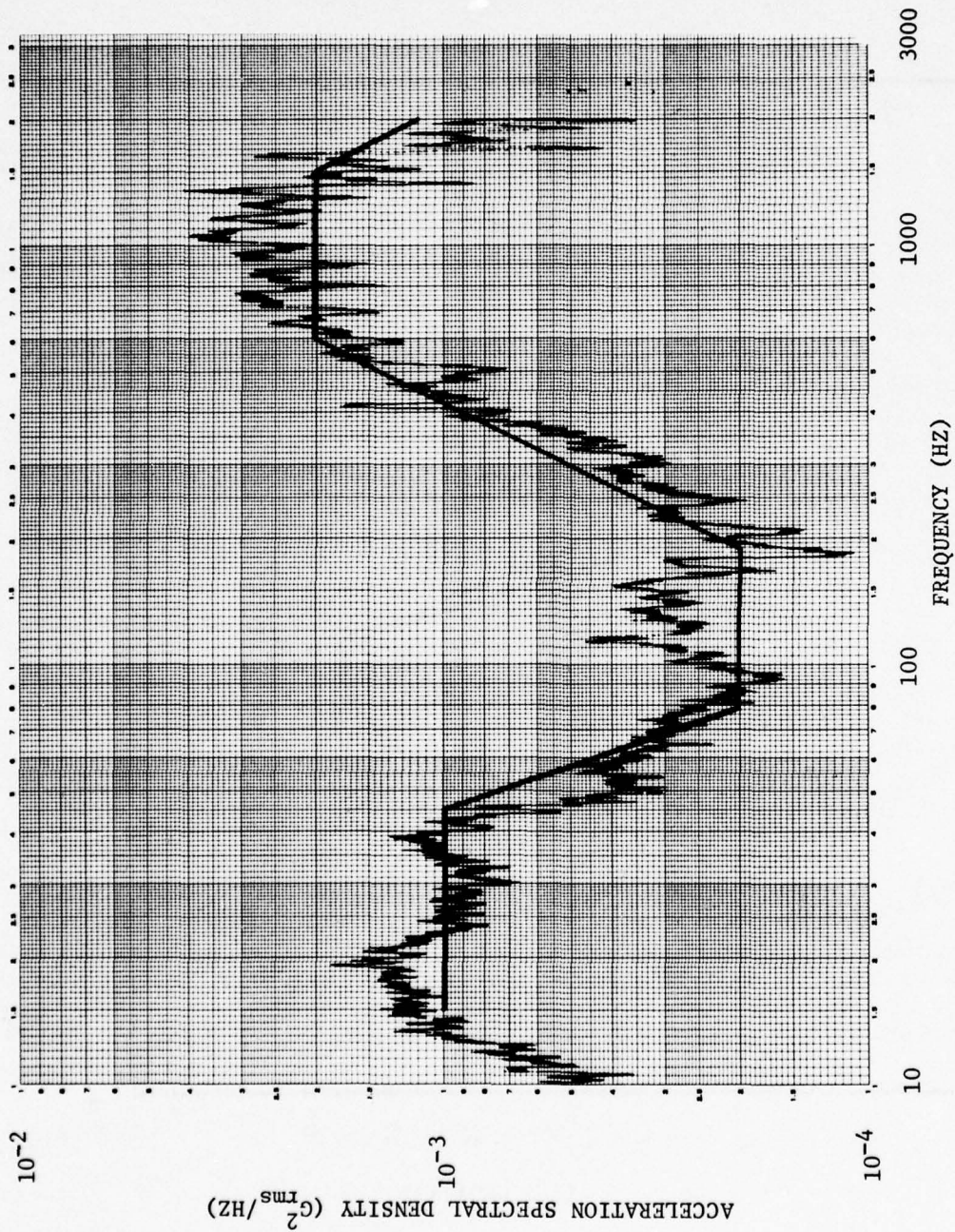
Figure 25.   Control of a Random Vibration Spectrum

SECTION IV

INITIAL OPERATIONAL EXPERIENCE

After checkout of the CERT facility via manual control, immediate testing of an airborne radar was initiated in the interest of evaluating the CERT concept. After seventy hours under manual operation, automatic control was installed and testing of the radar equipment resumed. Accumulated test time at the date of writing was approximately 670 hours. The major portion of this time was accrued by operating the facility 24 hours per day for five days per week. Only two facility breakdowns occurred during this period. The failed equipment consisted of a previously used solenoid valve for the LN_2 heat exchanger and the electromagnetic vibrator. The vibrator failure was caused by condensation of humid chamber air transferred to the vibrator by the pressure equalization line. This problem was resolved by placing a dessicant filter in this line.

Principal lessons learned relate to recognition of the need for simpler test operation and automatic facility monitoring features. It is recommended, for example, that a recorded voice channel on the magnetic tape deck be provided whereby specific test instructions (such as avionics test item on or off, vibration on or off, etc.) are announced automatically during test progress. Although continuous parameter monitoring is accomplished by plotting the controlled environmental parameters on X-Y plotters against preplotted profiles, protection against

excessive parameter deviation from setpoint depends upon the attentive-
ness of the monitoring technician. Therefore, it is also recommended
that all environmental parameters have deviation from setpoint alarms
and high and low range alarms.

The preliminary results of the first CERT test have been very
promising with respect to duplication of field failure modes and rates.
Specifically, several environmentally induced, intermittent failure
modes have been observed which would not have been found in current
reliability tests nor discovered by flight line technicians after air-
craft landing. Complete details of this initial CERT test are being
reported in a separate document (Reference 14).

SECTION V

CONCLUSIONS

Using Category II flight test data as a basis, dynamic environmental test profiles were constructed for conducting combined environments reliability testing (CERT) of avionics equipment in the laboratory. The dynamic environment profiles were employed in turn to define performance goals for a test facility in which CERT evaluations could be conducted. The successful development of an experimental CERT facility satisfying the basic performance goals is documented in this report.

Principal conclusions resulting from the design and development effort are:

(1) In the approach to CERT facility development adopted, it proved feasible to upgrade a limited capability temperature-altitude-vibration test chamber to the full spectrum of environments, parameter ranges, and dynamic response required for CERT testing and for a 1.5 KW cooling load.

(2) Advantages of an automated facility were amply demonstrated in the present development program in which an initially manually operated version of the facility requiring four operators was converted to automatic, magnetic tape controlled operation requiring the supervision of only a single attendant.

(3) Preliminary test experience with avionics equipment revealed an encouraging degree of correlation with field failure experience.

SECTION VI

RECOMMENDATIONS

On the basis of experience with the experimental facility described in this report, seven recommendations are made for improving the performance and economics of subsequent generations of CERT facilities.

1. An improvement in the airflow temperature response at low flow rates was found to be desirable. This may be accomplished by reducing, to the maximum extent possible, the thermal capacity of ducting connecting the thermal conditioning heat exchangers with the test equipment inlet. Approaches for accomplishing this could consist of:

   (1) shortening the length of piping

   (2) use of lighter gage ducting (tubing rather than pipe)

   (3) use of materials with low specific heat capacity

   (4) internal insulation within the ducts

   (5) concentric (duct within a duct) ducts with outer passage ventilated by conditioning heat exchanger heatant and coolant effluent streams up to the mixing point and by air of the desired temperature over the balance of the delivery duct length.

2. An improvement in the technical approach to providing the simulated static compartment air temperature is deemed desirable from the standpoint of accelerating response and improving the economics. As a means of saving the energy expense involved in conditioning the massive altitude chamber cold wall, an alternative ventilating approach to static compartment air conditioning is recommended. This would involve the location of a low heat capacity shroud surrounding the airborne equipment, while supplying a constant independent flow of air of appropriate temperature and humidity schedules to the space within the shroud which is then exhausted from the chamber by the same vacuum pump handling the simulated forced cooling air flow. An increase in exhauster capacity or decrease in upper altitude limit would be entailed. An additional complement of the temperature and humidity conditioning and control hardware would also be required.

3. Future systems should be designed to higher altitude capabilities such as 65,000 ft rather than the 38,000 ft of this experimental facility.

4. Substantial economic savings in acquisition and operating expense could result from pressurization of the process and control air to no more than the minimum feasible value consistent with the conceptual approach of the CERT facility implemented here. The 100 psig pressure used was an unavoidable constraint imposed by selection of an on-hand compressor.

This optimum pressure which is estimated to be about 30 psig should be determined from trade analyses involving mainly the type of dryer selected.

5. It is recommended that the handling and shipping environments be simulated through the inclusion of appropriate shocks in the vibration program.

6. To ensure that avionics equipment is realistically exercised during CERT tests, consideration should be given to the simulation of electrical power supply transients.

7. It is recommended that design of future CERT facilities include a controls and stability analysis if they depart from the decoupled parameter controls used in the present system and to determine whether additional compensation may be required for humidity, for example.

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APPENDIX A

ANALYSIS OF CHAMBER EVACUATION/PRESSURIZATION

A.1 INTRODUCTION

This appendix presents an analysis of the pressure response of a CERT chamber with air through-flow.

A.2 EVACUATION

A chamber of volume, V, receives a mass inflow of rate, W_{IN}, while being evacuated by a positive displacement vacuum pump at a volumetric exhaust rate, Q_{OUT}. The mass balance for the chamber may be stated as

$$V \frac{d\rho}{dt} = W_{IN} - \rho Q_{OUT} \tag{A-1}$$

With constant mass inflow rate, W_{IN}, and volumetric exhaust rate, Q_{OUT}, the density, ρ, of air in the chamber varies with time according to

$$\frac{\rho}{\rho_0} = \left(1 \frac{W_{IN}}{\rho_0 Q_{OUT}} \right) e^{-\frac{Q_{OUT}}{V} t} + \frac{W_{IN}}{\rho_0 Q_{OUT}} \tag{A-2}$$

where ρ_0, the T_0, P_0 are values of density, temperature and pressure at zero time.

With T, representing the chamber air temperature, the chamber pressure may be obtained from the density using the perfect gas law below.

$$P = \rho RT \tag{A-3}$$

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Equation A-2 becomes with the help of A-3

$$\frac{P}{P_0} = \frac{T}{T_0} \left[\frac{W_{IN}}{\rho_0 Q_{OUT}} + \left(1 - \frac{W_{IN}}{\rho_0 Q_{OUT}} \right) e^{-\frac{Q_{OUT}}{V} t} \right] \tag{A-4}$$

Based on equation A-4, the following conclusions are made.

(1) The pressure may be made to increase or decrease by adjusting the value of the ratio $W_{IN}/\rho_0 Q_{OUT}$ relative to unity. Values greater than unity result in increasing pressure while values less than unity cause pressure to decrease.

(2) The final value of chamber pressure, in addition to temperature, depends on the ratio $W_{IN}/\rho_0 Q_{OUT}$.

(3) The rate of pressure change depends on Q_{OUT}/V, the ratio of volumetric evacuation rate to chamber volume, in addition to the ratio $W_{IN}/\rho_0 Q_{OUT}$.

(4) The time constant (time to reach 63% of the change) is determined by Q_{OUT}/V only, and is equal to V/Q_{OUT}.

If initial and final values of pressure are denoted by P_0 and P_F, respectively, two additional useful expressions employed to make calculations in the text, may be derived:

$$P = P_0 \frac{T}{T_0} \left[\frac{P_F}{P_0} + \left(1 - \frac{P_F}{P_0} \right) e^{-t/\tau} \right] \tag{A-5}$$

$$\dot{P} = -\frac{(P - P_F)}{\tau} \tag{A-6}$$

A.3 ANALYSIS OF CHAMBER SIMULATED DIVE RATES

The simulation of altitude descent rates involves flow of ambient air into the chamber through an altitude control valve. The variable flow depends both on pressure ratio and valve opening. To analyze this situation equation A-1 may be rewritten as follows

$$V \frac{d\rho}{dt} + \rho \, Q_{OUT} = W_{IN_{MAX}} ; \; P/P_1 \le .5$$

$$V \frac{d\rho}{dt} + \rho \, Q_{OUT} = W_{IN_{MAX}} \sqrt{\frac{4}{3} \left(1 - \left(\frac{P}{P_1} \right)^2 \right)} \tag{A-7}$$

for $P/P_1 \ge .5$

The right-hand side is obtained from the Fluid Controls Institute valve sizing formula

$$\frac{W}{\rho_0} = 1360 \, C_v \sqrt{\frac{\Delta P}{T}} \sqrt{\frac{P_1 + P_2}{2}} \; ; \; P_2 \ge \frac{P_1}{2} \tag{A-8}$$

where ρ_0 is air density at ambient conditions and where it is seen that flow is related to $(P_1^2 - P_2^2)$.

Flow through valves is a throttling process and therefore an isothermal process. Accordingly from the perfect gas law

$$\rho = \frac{P}{RT_0} \tag{A-9}$$

It is advantageous to nondimensionalize equation A-7 in preparation for integrating. Thus with

$$X = \frac{P}{P_o} \ , \ \tau = \frac{V}{Q_{OUT}} \ , \ t^+ = \frac{t}{\tau} \ , \qquad (\tau \text{ is the time constant})$$

and equation A-9 we obtain

$$\frac{dX}{dt^+} + X = \frac{W_{MAX} \ \tau}{\rho_o \ V} \ \sqrt{\frac{4}{3} (1 - X^2)} \ ; \ X \geq .5 \qquad (A-10)$$

It is clear from equation A-10 that the steady-state pressure ratio is less than unity. In fact with

$$G = \frac{2 \ W_{MAX}}{\sqrt{3} \ \rho_o \ V} \quad \text{and} \quad \frac{dX}{dt^+} = 0,$$

it may be determined that

$$X_{SS} = \sqrt{\frac{G^2}{1 + G^2}} \qquad (A-11)$$

The solution to equation A-10 is obtained by evaluating the following integral

$$t^+ = \int_{X_1}^{X} \frac{d \xi}{\frac{\sqrt{3}}{2} G = \xi} \ ; \ X_1 \leq X \leq .5$$

$$(A-12)$$

$$t^+ = \int_{X_1}^{.5} \frac{d \xi}{\frac{\sqrt{3}}{2} G - \xi} + \int_{.5}^{X} \frac{d \xi}{G\sqrt{1 - \xi^2} - \xi} \ ; \ .5 \leq X \leq X_{SS}$$

Since the denominator of the second integrand has a singularity at ξ = X_{SS}, numerical values of t^+ will grow without limit as X approaches its steady-state value. The integration is therefore carried out only to some fraction of X_{SS}. Here we use X_{final} = .9 X_{SS}. For the purpose of integrating equation A-12 it may be noted that

$$\lim_{X \to X_{SS}} \frac{1}{G\sqrt{1 - X^2} - X} = \frac{1}{(G^2 + 1)(X_{SS} - X)} \tag{A-13}$$

Therefore the integrand will be approximated by

$$\frac{1}{G\sqrt{1 - X^2} - X} = \frac{1}{(G^2 + 1)(X_{SS} - X)} + \sum_{i=0}^{n} C: (X - 5)^i \tag{A-14}$$

Figure 26 shows the excellent approximation to the left side obtained by the expression on the right side of the following formula.

$$\frac{1}{G\sqrt{1 - X^2} - X} - \frac{1}{(1 + G^2)(X_{SS} - X)} = .185 + .110(X - .5)$$
$$+ 8.82(X - .5)^5 \tag{A-15}$$

when X_{SS} = .9796 and G = 24.76.

This corresponds to P_2 = 14.4 psia, P_0 = 14.7 psia. Accordingly the integrals in equation A-12 are evaluated as follows

$$t^+ = \ln \left[\frac{\frac{\sqrt{3}}{2} G - X_1}{\frac{\sqrt{3}}{2} G - .5} \right] + \frac{1}{1 + G^2} \ln \left[\frac{X_{SS} - .5}{X_{SS} - .9X_{SS}} \right]$$

$$\tag{A-16}$$

$$+ \sum_{i=0}^{n} \frac{C_i}{(i - 1)} (.9X_{SS} - .5)^{i+1} \; ; \text{ where } X_1 < .5$$

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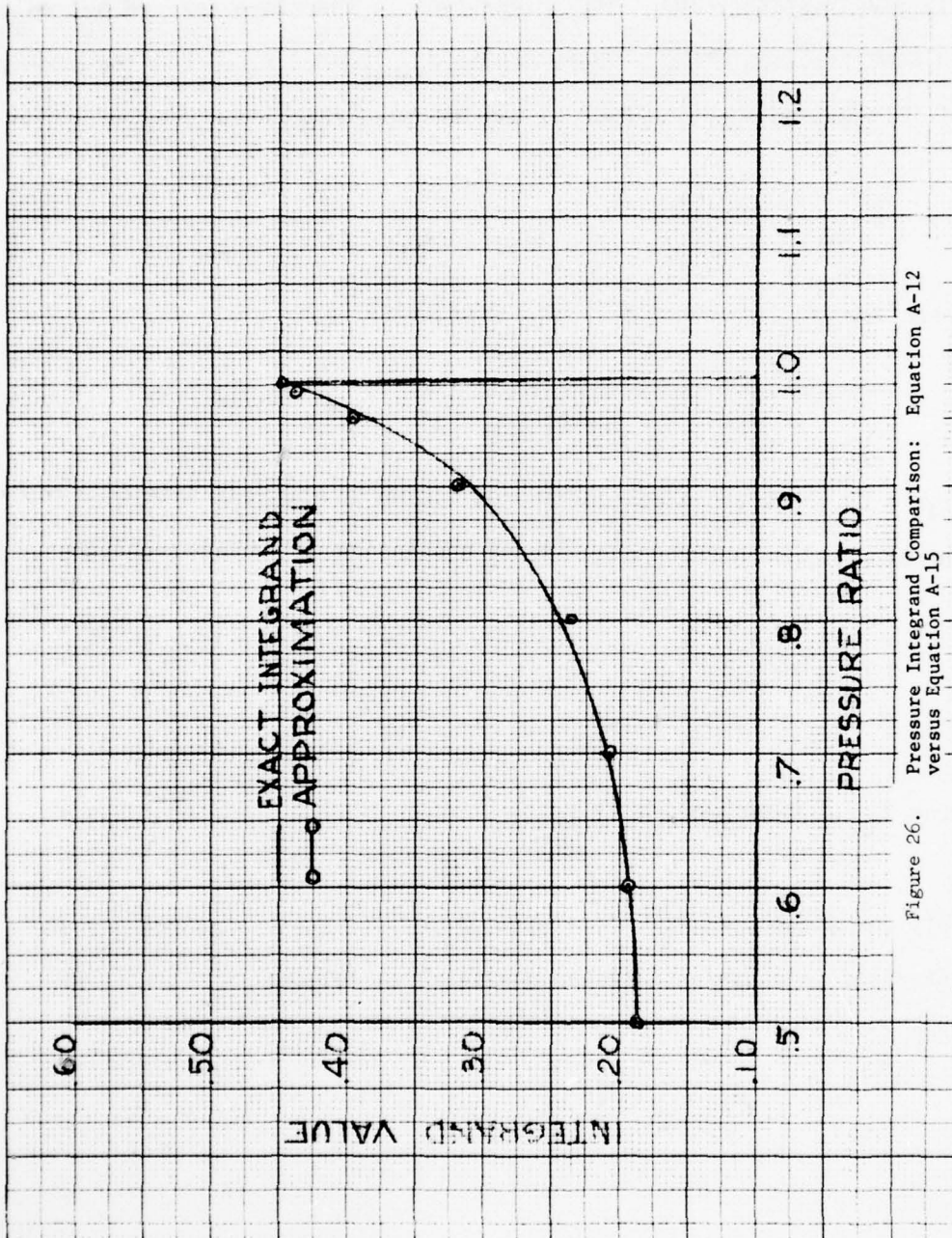
Figure 26. Pressure Integrand Comparison: Equation A-12 versus Equation A-15

For the situation to which equation A-15 applies, A-16 was evaluated to give

$$t^+ = .02156 + .06415 + .0706 + .008011 + .004542$$

$$t^+ = .16886 \qquad\qquad (A-17)$$

$$t = t^+ \tau = .169 \times 12 = 2.03 \text{ secs.}$$

APPENDIX B

TRANSIENT TEMPERATURE RESPONSE OF AIR

FLOWING THROUGH A DUCT

B.1 INTRODUCTION

The temperature response of air emerging from a duct will lag be-
hind its instantaneous inlet temperature due to (1) transport delay
which is usually small except for very long ducts, and (2) a relaxation
time associated with establishing steady-state conditions in the duct
wall. The relationship between duct inlet temperature and exit temper-
ature variations is derived in this appendix. For simplicity the duct
is considered well insulated from its external environment. The duct
wall has an average temperature T_w.

B.2 DISCUSSION

Reference should be made to the definition sketch below:
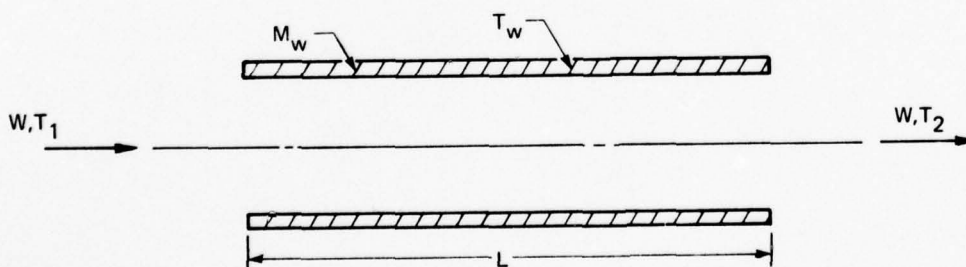
The air exchanges heat with the duct wall according to the heat ex-
changer equation

$$T_1 - T_2 = \varepsilon(T - T_w) \tag{B-1}$$

The rate of wall temperature change is proportional to the heat loss obtained by the air in accordance with

$$M_w C \frac{dT_w}{dt} = WC_p (T_1 - T_2) \tag{B-2}$$

Applying the Laplace transform to equations B-1 and B-2 and solving simultaneously for the transfer function between inlet and exit temperatures yields:

$$\frac{\overline{T}_1}{\overline{T}_2} = \frac{\dfrac{1}{1 - \varepsilon} \dfrac{(S + WC_p \varepsilon)}{M_w C}}{\dfrac{S + WC_p}{M_w C} \left(\dfrac{\varepsilon}{1 - \varepsilon}\right)} \tag{B-3}$$

The time constant for the transient behavior is seen to equal

$$\tau = \frac{M_w C(1 - \varepsilon)}{WC_p \ \varepsilon} \tag{B-4}$$

Assuming a second duct or valve to be inserted into the flow circuit, the transfer function between inlet and exit may be obtained by repeated application of equation B-3

$$\frac{\overline{T}_1}{\overline{T}_3} = \frac{\overline{T}_1}{\overline{T}_2} \ \frac{\overline{T}_2}{\overline{T}_3} \tag{B-5}$$

or letting $\gamma = \dfrac{WC_p \varepsilon}{M_w C}$ and $a = \dfrac{\gamma}{1 - \varepsilon}$

$$\frac{\overline{T}_1}{\overline{T}_3} = \left(\frac{1}{1 - \varepsilon_1}\right) \left(\frac{1}{1 - \varepsilon_2}\right) \left(\frac{S + \gamma_1}{S + a_1}\right) \left(\frac{W + \gamma_2}{S + a_2}\right) \tag{B-6}$$

If a ramp temperature change is desired at the duct outlet, then

$$T_3 = \frac{\Delta T}{\tau} t$$

or

$$\bar{T}_3 = \frac{\Delta T}{\tau} \frac{1}{S^2}$$

(B-7)

The inlet temperature variation needed to maintain this outlet rate of change is given by inverting equation B-6

$$T_1 = \frac{1}{1 - \epsilon_1} \cdot \frac{1}{1 - \epsilon_2} \frac{\Delta T}{\tau} \left[A_1(t) + (\gamma_1 + \gamma_2) A_2(t) \right.$$

$$\left. + \gamma_1 \gamma_2 A_3(t) \right]$$

(B-8)

where
$$A_1(t) = \frac{1}{a_1 - a_2} \left(e^{-a_1 t} - e^{-a_2 t} \right)$$

$$A_2(t) = \frac{1}{a_1 - a_2} \left[\frac{1 - e^{-a_1 t}}{a_1} - \frac{1 - e^{-a_2 t}}{a_2} \right]$$

$$A_3(t) = \frac{1}{a_1 - a_2} \left[\frac{t}{a_1} - \frac{t}{a_2} - \frac{\left(1 - e^{-a_1 t}\right)}{a_1^2} + \frac{\left(1 - e^{-a_2 t}\right)}{a_2^2} \right]$$

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The solution, equation B-8, has been programmed for a digital computer to evaluate the entrance temperatures required to provide various temperature change rates at the duct exit with and without valves in the line. Some of the results are given in Table II. In plain evidence is the conclusion that short, small diameter, thin walled tubing is desirable to minimize the attenuation of temperature signals in passage through a duct, while valves should definitely not be located between mixing and delivery points. Table II presents the inlet temperature rise or drop from an initial steady-state value after the lapse of 2.65 minutes that would be required to change the exit air temperature at a constant rate of 100°F per minute. The two cases considered here include various sizes of 3-foot long lengths of standard copper tubing with and without a 16-lb valve present in the line.

A program listing appears in Appendix C. The program, called CERT, allows flow rates, ducting lengths, and temperature change rates and time to be varied in addition to those reflected in the sample cases of Table II.

The ostensible limitation to perfectly insulated ducts may be relaxed by modifying wall thickness or heat capacity of the duct to account for heat storage in the duct insulation.

APPENDIX C

LISTING OF COMPUTER PROGRAM "CERT"

```
      PROGRAM CERT(INPUT,OUTPUT)
C COMPUTES TEMP UPSTREAM OF DUCT AND VALVE TO GIVE REQD RAMP TEMP OUTPUT
C W IS MASS FLOW RATE OF AIR, LBM/MIN
C RHOW IS MASS DENSITY OF DUCT WALL MATERIAL, LBM/FT**3
C DELT IS WALL THICKNESS, INCHES
C VM IS MASS OF VALVE, LBM
C TEE IS TIME, MIN
C  TAU IS ONE MINUTE
C  TEMP IS TEMP CHANGE IN ONE MINUTE
C  D(1) IS DUCT DIAMETER , IN
C  D(2) IS VALVE DIAMETER, IN
C  TL(1) IS DUCT LENGTH, FT
C C(1) IS SPECIFIC HEAT OF DUCT, BTU/LBM/DEG F
C C(2) IS SPECIFIC HEAT OF VALVE, BTU/LBM/DEG F
      DIMENSION D(3),TL (3),C(3),RE(3),H(3),S(3),TU(3),EPS(3),WT(3),GAM(3
     1),8(3)
    2 READ 5, W,TEE, TEMP,RHOW, DELT, VM
    5 FORMAT (6F10.6)
      IF (W.EQ.0.0) GO TO 20
      READ 5, (D(I),TL(I),C(I), I=1,2)
      PRINT 6
    6 FORMAT (3X,1HW,9X,3HTEE, 7X,4HTEMP,6X,4HRHOW,6X,4HDELT,6X,2HVM/)
      PRINT 5, W,TEE, TEMP,RHOW,DELT, VM
      PRINT 7
```

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```
  7 FORMAT (3X,2HD1,8X,3HTL1,7X,2HC1,8X,2HD2,8X,3HTL1,7X,2HC2/)
    PRINT 5, (D(I),TL(I),C(I), I=1,2)
    PI= 3.14159
    VMU=.0415
    TAU=1.
    DO 10 I=1,2
    RE(I)=48./PI*60./VMU*W/D(I)
    PRINT 8, RE(I)
    H(I)=.023*.87 3*RE(I)**.8*.0144/D(I)*12.
    IF (I.EQ.2) H(I)=1.5*H(I)
    PRINT 8, H(I)
    S(I)=PI/12.*D(I)*TL(I)
    PRINT 8, S(I)
    TU(I)=H(I)/W*S(I)/.24*1./60.
    PRINT 8, TU(I)
    EPS(I)=1.-EXP(-TU(I))
    PRINT 8, EPS(I)
    WT(I)=RHOW*S(I)/12.*DELT
    WT(2)=VM
    PRINT 8, WT(I)
    GAM(I)=.24/WT(I)*W/C(I)*EPS(I)
    PRINT 8, GAM(I)
    B(I)= GAM(I)/(1.-EPS(I)
    PRINT 8, B(I)
 10 CONTINUE
    A1=(EXP(-B(1)*TEE)-EXP(-B(2)*TEE))/(B(2)-B(1))
```

```
      PRINT 8, A1
      A2=((1.-EXP(-B(1)*TEE))/B(1)-(1.-EXP(-B(2))*TEE))/B(2))/B(2)-B)1))
      PRINT 8, A2
      A3=(TEE*(1./B(1)-1./B(2))-(1.-EXP(-B(1)*TEE))/B(1)**2+(1.-EXP(-B(2
     1)*TEE))/B(2)**2)/(B(2)-B(1))
      PRINT 8, A3
      T1=TEMP/TAU*(A1+(GAM(1)+GAM(2)*A2+GAM(1)*GAM(2)*A3)/((1.-EPS(1))*




     1(1.-EPS(2)))
    8 FORMAT (5X,1E15.7)
      PRINT 9, T1
    9 FORMAT (5X,4HT1=   ,E15.7)
      E1=(1.-EXP(-B(1)*TEE))/B(1)
      E2=GAM(1)*(TEE-E1)/B(1)
      TNV=TEMP/TAU*(E1+E2)/(1.-EPS(1))
      PRINT 19, TNV
   19 FORMAT (5X,4HTNV=,E15.7)
      GO TO 2
   20 STOP
      END
```

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